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President, A. A. HOPPE
Vice-President, R. G. DOLAN
Secretary-Treasurer, S. L. ROLLAND
Board of Governors: R. E. PAULING, R. L. PATTON, EARLE W. GRAY

Ontario

Headquarters, Toronto, Ont.
Meets: First Monday

President, H. D. HENION
Vice-President, J. W. O'NEILL
Secretary-Treasurer, H. R. ROTH
Board of Governors: C. TASKER, F. E. ELLIS, J. H. FOX

Oregon

Headquarters, Portland, Ore.
Meets: Thursday after first Tuesday

President, J. DONALD KROEKER
Vice-President, T. E. TAYLOR
Secretary, C. E. HEINKEL
Treasurer, H. B. NIELSEN
Board of Governors: E. E. CARROLL, C. M. MACGREGOR, F. H. URBAN

Pacific Northwest

Headquarters, Seattle, Wash.
Meets: Second Tuesday

President, R. O. WESLEY
Vice-President, M. J. HAUAN
Secretary, H. T. GRIFFITH
Treasurer, E. E. FOOTE
Board of Governors: S. D. PETERSON, W. W. COX, C. W. MAY

OFFICERS OF LOCAL CHAPTERS—1939 (*continued*)

Philadelphia

Headquarters, Philadelphia, Pa.
Meets: Second Thursday

President, R. F. HUNGER
First Vice-President, C. B. EASTMAN
Second Vice-President, H. B. HEDGES
Secretary, EDWIN ELLIOT
Treasurer, H. H. MATHER
Board of Governors: R. F. HUNGER, H. H. ERICKSON, C. B. EASTMAN, H. B. HEDGES, H. H. MATHER, EDWIN ELLIOT, M. G. KERSHAW

Pittsburgh

Headquarters, Pittsburgh, Pa.
Meets: Second Monday

President, R. A. MILLER
Vice-President, F. C. MCINTOSH
Secretary, T. F. ROCKWELL
Treasurer, L. S. MAEHLING
Board of Governors: J. F. COLLINS, JR., E. H. RIESMEYER, JR., E. C. SMYERS

St. Louis

Headquarters, St. Louis, Mo.
Meets: First Tuesday

President, R. J. TENKONOHY
First Vice-President, C. E. HARTWEIN
Second Vice-President, D. J. FAGIN
Secretary, C. F. BOESTER
Treasurer, M. F. CARLOCK
Board of Governors: R. J. TENKONOHY, C. E. HARTWEIN, D. J. FAGIN, C. F. BOESTER, M. F. CARLOCK, E. E. CARLSON, G. B. RODENHEISER, F. C. LAUFKETTER, J. H. CARTER

South Texas

Headquarters, Houston, Tex.
Meets: Third Friday

President, R. E. TAYLOR
Vice-President, R. E. CHRONE
Secretary, W. H. BADGETT
Treasurer, I. E. ROWE
Board of Governors: J. A. KIESLING, C. A. MCKINNEY, R. W. KOTZEBUE

Southern California

Headquarters, Los Angeles, Calif.
Meets: Second Tuesday

President, J. F. PARK
Vice-President, H. H. DOUGLAS
Secretary, E. P. WELLS
Treasurer, GLEN MILLER
Board of Governors: H. M. HENDRICKSON, A. J. HESS, W. D. FABLING, O. W. OTT

Washington, D. C.

Headquarters, Washington, D. C.
Meets: Second Wednesday

President, T. H. URDAHL
Vice-President, F. E. SPURNEY
Secretary, P. H. LOUGHRAN, JR.
Treasurer, G. R. WALZ
Board of Governors: E. V. FINERAN, S. P. EAGLETON, S. L. GREGG

Western Michigan

Headquarters, Grand Rapids, Mich.
Meets: Second Monday

President, B. F. MCLOUTH
Vice-President, T. D. STAFFORD
Secretary, C. H. PESTERFIELD
Treasurer, C. H. MORTON
Board of Governors: C. R. MCCONNER, W. G. SCHLICHTING, F. C. WARREN

Western New York

Headquarters, Buffalo, N. Y.
Meets: Second Monday

President, L. P. SAUNDERS
First Vice-President, C. A. GIFFORD
Second Vice-President, W. R. HEATH
Secretary, H. C. SCHAFER
Treasurer, B. C. CANDEE
Board of Governors: D. J. MAHONEY, M. C. BEMAN, JOSEPH DAVIS, ROSWELL FARNHAM, J. J. LANDERS

Wisconsin

Headquarters, Milwaukee, Wis.
Meets: Third Monday

President, D. W. NELSON
Vice-President, H. C. FRENTZEL
Secretary, T. M. HUGHEY
Treasurer, A. S. KRENZ
Board of Governors: J. H. VOLK, E. A. JONES, J. S. JUNG

TRANSACTIONS
of
AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

No. 1104

FORTY-FIFTH ANNUAL MEETING, 1939

MEMBERS who attended the 45th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS found the technical sessions intensely interesting and the hospitality of Pittsburgh Chapter most enjoyable. It was a notable meeting from the standpoint of attendance, interesting discussions, the variety of technical papers, and the fact that 1939 commemorated the establishment of the Society's Research Laboratory in Pittsburgh. It was entirely fitting that this meeting should be held in Pittsburgh, and the majority of men who had served as chairmen of the Committee on Research since 1919 were present. The total registration was 570 with more than 300 members registered.

The 45th Annual Meeting of the Society was opened by Pres. E. Holt Gurney, Toronto, Ont., at the William Penn Hotel, Pittsburgh, January 23.

President Gurney read the annual report of the president.

President's Address

The general activities of the Society are covered in the report of your Council; of the Research Committee by the report of its chairman; and the financial situation is clearly set forth by the chairman of the Finance Committee. I regard progress in all these matters this year as reasonably satisfactory. There are, however, certain matters which I think it is more appropriate to discuss with the membership direct in Annual Meeting rather than with the Council.

In the first place, I believe that we should get away from the tradition that the president should visit most of the Chapters, or all of them. I am very conscious of the fact that, as I write this, I am far better qualified to be president of the Society than when I began my term. I visited some 16 Chapters this year, in addition to the seven that I visited last year. Thus, I came to my term of office without a knowledge of the men active in the Chapters which I now possess. If your president had had the advantage of this knowledge when he appointed the committees for this

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year, several able men, actively interested in the work of the Society, would have had a larger part in it through membership in certain committees.

Further, while I have spoken to several outside bodies, I have had to refuse many more invitations to do this sort of thing than I have accepted, just because of the pressure on your president to visit Chapters. As the Society gains in authority, and especially as we have larger contact with the medical profession, I consider that it will be more and more important that your president be free to do this type of speaking rather than to the Chapters.

Our Constitution sets out that the president "shall visit as many of the Chapters as his convenience permits." I believe this should be changed to read, "The president, the first vice-president and the second vice-president shall visit as many of the Chapters as their convenience permits." But I am perfectly clear that, as a matter of practice, most of these Chapter visits should be made by the first and second vice-presidents. If this is done, they will better discharge their duties, when they reach the chair, and the president will be free to make very important outside contacts.

I am not of opinion that this modification of policy should be effected in one year; this would be unwise. I believe that this year, for example, your president should certainly go to the Pacific Coast, and the South; our members in these far away sections are of necessity less represented here than nearer parts of the country. I believe that this year your president should explain, wherever he goes, what is contemplated. The following year the policy could be pushed further, and it could be finally worked out over a period, say of three years. Meanwhile, the other important contacts that the president should make annually can gradually be formed.

I believe that it is very important that more of the money spent by the Speakers' Bureau should be spent in getting outstanding men for the far-away Chapters, and that less should be done for those Chapters in the great metropolitan centers of the east, especially where they are in a university town. Your Speakers' Bureau has a very difficult problem. Again and again an outstanding speaker will consent to leave his work for two nights, and this means that the Chapters who need help least have benefited in largest measure from the activities of your Speakers' Bureau. I would like to see the Speakers' Bureau finance a trip to the coast for two or three outstanding men. They could go out by Omaha, Oklahoma and Texas, and return via Winnipeg. They might consider this as a spring or fall holiday, if their expenses were paid for the trip, and I believe such a move would do a great deal to bridge the distance and the *east versus west* feeling inevitable in any great national organization, whether it be in Canada or the United States. I am sure this would make for greater solidarity in the ASHVE.

Finally, and to me much the most important item of all, is your procedure in selecting Council and officers. Under our present set-up a Nominating Committee meets, consisting of one representative from each Chapter. There has arisen, inevitably, under this situation, a feeling that it is the *turn* of a given district or a given Chapter to be represented. This has consequently meant that a man who has no Chapter affiliations is rather unlikely to reach office. Nearly 50 per cent of our fraternity have no Chapter affiliations, and many of them cannot have one because of the geography of the situation. Under this set-up, there is some question in my mind, whether, for example, Dr. Willard could ever have reached the presidency. I think of two or three outstanding men in the United States whom I hope to see some day in the chair of this Society, but who in their present location, find it unnatural to be affiliated with a Chapter.

Further, it seems to me that the selection of Council men each year should always keep a fair distribution as between the three great classes that make up our Society. I refer to the executive business man in engineering work, the professional engineer, and the academic man. These three groups intermingle in our Society in a way that gives so much of our strength; but when the main consideration becomes geographical, this very important phase of the matter is apt to be overlooked.

In all my associations with things that have to be done, it has seemed to me that my most important job has always been the question of personnel; yet I am now retiring from active leadership in this Society without any voice whatever as to who may come on this board in the future, or who may go on to office. It seems

to me almost a truism that I must know more about this than any man who is sent here to represent a Chapter. I know this is so, because for the last two years my opinion has been sought by members of your Nominating Committee, and it is particularly so this year after my extensive travels in your behalf.

I suggest two things: first, that your immediate past president shall be ex-officio a member of the Nominating Committee, and that this be given effect to at once; second, that you appoint a committee to study this whole situation. I would suggest the president of your Advisory Council, the president of your Society, the chairman of your Nominating Committee, the chairman of your Constitution and By-Laws Committee, and, if you like, two men appointed from the floor of this convention. Surely such a group will suggest nothing which will militate against a democratic set-up. I am altogether in sympathy with the proposition that no officers of this Society shall find themselves in a self-perpetuating position; but most certainly we must do something to qualify a position in which 25 Chapter representatives meet to select our officers and directors without knowledge of the men who cannot belong to a Chapter, and without the especial knowledge which comes to a senior officer in this organization after one, two or three years' travel in your behalf.

I have difficulty in finding words to express my great appreciation of the complete and enthusiastic cooperation I have had from everybody throughout my year, and I also wish to pay tribute to the men and women in the employ of the Society for both their ability and their remarkable loyalty.

All of which is respectfully submitted.

E. HOLT GURNEY, *President.*

The secretary of the Society, A. V. Hutchinson, New York, read the report of the Council and the report of the secretary.

Report of the Council

During 1938 meetings of the Council were held in New York City, Buffalo, Hot Springs, Montreal and Pittsburgh. On January 28, 1938, the Council was organized and Council committees were appointed by Pres. E. Holt Gurney. Special committees required by the Constitution and By-Laws were organized; depositories for the Society's funds were selected, and a budget of $106,600 was adopted.

A. V. Hutchinson was appointed secretary; the firm of Tusa and La Bella was retained as certified public accountant to audit the Society's books. A fund of $5,000 was transferred to the Committee on Research by the Council. New offices for the Society's headquarters were selected at 51 Madison Ave., and a two-year lease was negotiated with renewal option for another year.

A technical secretary of the Society was appointed, John James, and arrangements were made for him to assist the chairman of the Committee on Research in administrative work. A plan of reorganization of activities was studied by the Council and a campaign of educational publicity was inaugurated. Authorization for the disposal of surplus publications of the Society was given and members were offered the first opportunity to purchase back numbers of the JOURNAL and TRANSACTIONS at nominal prices.

The Exchange Service Plan between the Society and the *ASRE* was renewed. Charters were authorized by the Council for three new chapters: the North Texas and South Texas Chapters and the North Carolina Chapter.

The Semi-Annual Meeting place for 1939 was selected at Mackinac Island, and the dates of the meeting are July 4, 5 and 6. The meeting place for the 46th Annual Meeting was chosen—Cleveland, Ohio, and the meeting will be held during the fourth week in January, 1940.

Some of the special projects studied by committees during the year were research reorganization, engineering and public health, air conditioning legislation, Society standards, educational publicity, regional meetings, engineering fees for housing

projects, procedure for the guidance of the Admission and Advancement Committee, and engineering scholarships.

The Council carried out its functions as required by the Constitution and By-Laws in nominating candidates to serve on the Committee on Research; voted on nominations for the F. Paul Anderson Medal; approved Candidates for Membership, and authorized resignations and cancellations of memberships.

The Council obtained legal opinion on the Society's activities under its charter, Constitution, By-Laws and Rules. It approved programs for Annual and Semi-Annual Meetings and carried on all routine activities specified under the Society's regulations.

Respectfully submitted,

THE COUNCIL.

Report of the Secretary

A detailed account of the daily operations carried on in the headquarters office in the calendar year 1938 would make a voluminous document and would include much statistical material that might prove to be very dry reading.

The main activities of the secretary's office deal with the work of the Council and special committees; production of THE GUIDE, JOURNAL, TRANSACTIONS; standards and codes; membership, which includes elections, advancements, cancellations, address changes, and dues collections; employment service; chapter activities; Speakers Bureau; sales promotion and publicity.

With the total membership approximately 3000 the work requires a staff of 10 people. A new activity was inaugurated this year with much of the routine administrative work on research transferred to headquarters and practically half of the time of the technical secretary was devoted to this work. This activity requires clerical and bookkeeping service and has been effective in relieving the chairman and director of research of much detailed work.

Detailed figures will be given in the various committee reports about the status of membership, the production of THE GUIDE, and the financial condition of the Society.

It is interesting to note that dues collections in 1938 just reached the 10 year average although unusual efforts were made to stimulate collections during the last quarter of the year. About 10 per cent of the membership now have one year's dues in arrears. In the survey of dues collections it was noticed that 60 per cent of total dues collected in any calendar year comes in during the first four months and the balance comes in in small percentages during each of the other months.

In compiling the Roll of Membership to be published in THE GUIDE, it was noticed that nearly 70 per cent of the members had address changes. When the first request was sent out only 40 per cent of the members responded, which added considerably to the cost of compiling the list.

A favorable increase in membership was recorded during 1938 with 380 new applications filed, but the Admission and Advancement Committee on numerous occasions found it necessary to request additional information because of inadequate information on the original application.

It would be well for every member who signs an application as a proposer or seconder to see that complete information on the education and professional experience of the candidate is given. Personal signatures of sponsors are required and it is also important that a sponsor personally know the candidate, so that he can attest to his engineering ability.

A total of 12,600 GUIDES were printed and bound in 1938 and over 12,450 were distributed; THE GUIDE was mailed in all states of the United States, all of the provinces of Canada and in the majority of countries throughout the world. Reports indicate that parts of THE GUIDE have been translated into many foreign languages and data from THE GUIDE are to be found in most of the modern texts on air conditioning and in the courses offered by schools and colleges. If all of the copies

of the 1938 edition of THE GUIDE could be placed in a pile, it would be over 2½ times the height of the Empire State Building, and their weight would total 25½ tons. When it is visualized that each page of THE GUIDE is read at least four times before it gets into print and that, outside of the membership, shipments are being made in small quantities each day, there is a large amount of work involved in acknowledging orders, labeling, shipping and billing the complete edition.

Every effort is made at the headquarters office to render effective service to members and the staff is always on call for service.

Respectfully submitted,

A. V. HUTCHINSON, *Secretary.*

Report of the Finance Committee

1st Vice-Pres. J. F. McIntire, Detroit, Mich., chairman of the Finance Committee, gave the report of the Society's certified public accountant for the calendar year 1938. In commenting on the financial condition of the Society, Mr. McIntire said that dues collections were of considerable concern to the committee and that the Council found it necessary to drop members in arrears because of the expense involved in carrying men who did not pay up.

Report of Certified Public Accountant

January 17, 1939.

American Society of Heating and
Ventilating Engineers,
51 Madison Avenue,
New York, N. Y.

Gentlemen:

Pursuant to your request we made an examination of the books of account and records of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS—New York, N. Y., and the Research Fund for the calendar year ended December 31, 1938 and submit herewith our report.

The work covered a verification of the Assets and Liabilities as of the date previously stated and a review of the operating accounts for the year ended December 31, 1938. For the period recorded cash receipts were traced into the depositories; the cancelled bank vouchers were inspected and compared with the cash records and the Dues Income was accounted for.

Submitted herewith is a Balance Sheet showing the financial condition of the Society on December 31, 1938 and your attention is directed to the following comments thereon:

CASH

Cash on Deposit was verified by direct communication with the banks and reconcilement of the amounts reported to us with the balance shown by the books of the Society.

The Petty Cash kept in the New York Office was verified by count.

MARKETABLE SECURITIES

There is attached hereto a schedule of negotiable bonds which were verified by direct communication with the Bankers Trust Company where same are deposited for safekeeping. No adjustment has been made of the $3,862.96 shrinkage in the market value of these securities, same being included in the attached Balance Sheet at cost.

ACCOUNTS RECEIVABLE

A list of the Membership Dues Receivable as of December 31, 1938 furnished to us

by the management was checked to the individual ledger cards and found in agreement with the General Ledger Control. The unpaid dues may be summarized as follows:

Dues billed in 1938	$ 6,711.28
Dues billed in 1937	2,085.75
Dues billed in prior years	5,049.60
TOTAL	$13,846.63

Amounts due from GUIDE Advertisers and Other Accounts Receivable verified by trial balance of the individual ledger accounts were found in agreement with the General Ledger Control.

The Reserve for Dues Doubtful of Collection found on the books has been increased by $3,676.58 while the Reserve for advertisements and Sundry Receivables Doubtful of Collection has been decreased by $58.50. The adjusted Reserves now shown on the accompanying Balance Sheet in our opinion are ample to cover losses resulting from future realization.

INVENTORIES

Items appearing under this caption are based upon quantities submitted by the management and computations made by us. The Transactions Inventory priced at cost may be summarized as follows:

Prior Years	$1,750.00
1932	383.10
1933–34	652.00
1935–36	51.15
1937–38	395.00
TOTAL	$3,231.25

FURNITURE, FIXTURES AND LIBRARY

Furniture, Fixtures and Library are shown herein at the book values without appraisal by us; we did, however, provide for depreciation of Furniture and Fixtures at the rate of ten (10%) per cent per annum.

DEFERRED CHARGES

Items appearing under this caption have been computed by us.

ACCOUNTS PAYABLE

On December 31, 1938 there remained unpaid invoices amounting to $1,481.55; also the sum of $21,677.27 estimated by the management necessary to complete and make the first mailing of the 1939 GUIDE and Vols. 43 and 44 of TRANSACTIONS.

PAYABLE TO RESEARCH FUND

The amounts due the Research Fund representing 40% of Members and Associates Dues were determined from computations made by us in accordance with Section 5 Article 3 of the By-Laws.

ACCRUED ACCOUNTS

Additional compensation to the Secretary and the clerical staff of the Society has been computed in accordance with the instructions of the Finance Committee.

DEFERRED INCOME

Members Prepaid Dues were ascertained by trial balance of the individual ledger

cards. Prepaid Dues from proposed Members were verified from inspection of the applications found on file.

GENERAL FUND

There is attached hereto a schedule showing the changes that have occurred in the General Fund during the calendar year 1938.

SPECIFIC FUNDS

The specific funds are shown on the attached Balance Sheet after adjustment for interest earned.

Operations of the Research Fund for the calendar year 1938 have been detailed in a separate report rendered to you.

OPERATIONS

There is attached hereto Statements of Income and Expenses for the calendar year 1938 showing an excess of Expenses over Income from Society activities of $8,300.27 and a net income of $8,252.91 from GUIDE Operations. In the preparation of these statements 30% of the Salaries and Office expenses have been allocated to GUIDE Operations.

Respectfully submitted,

TUSA & LA BELLA,
(*Signed*) Frank G. Tusa,
Certified Public Accountants.

BALANCE SHEET

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
NEW YORK, N. Y.

December 31, 1938

ASSETS

SOCIETY			
CASH			
On Deposit	$42,517.50		
On Hand	100.00		
		$42,617.50	
In Closed Bank		418.97	
			$43,036.47
RESERVE FUNDS			
On Deposit		$ 1,587.18	
On Hand for Deposit		125.82	
			1,713.00
INVESTMENTS (AT COST)			
Securities (Market Value $19,284.58)		$19,171.14	
Add: Accrued Interest		61.46	
			19,232.60
ACCOUNTS RECEIVABLE			
Membership Dues	$13,846.63		
Less: Reserve for Doubtful	8,105.85	$ 5,740.78	
Advertisers and Sundry Debtors	$32,030.15		
Less: Reserve for Doubtful	1,462.62		
		30,567.53	
			36,308.31
INVENTORIES			
Transactions		$ 3,231.25	
Emblems and Certificate Frames		69.10	
Stationery		226.36	
			3,526.71
PERMANENT			
Library		$ 300.00	
Furniture and Fixtures	$ 6,979.90		
Less: Reserve for Depreciation	5,206.59	1,773.31	
			2,073.31

DEFERRED CHARGES			
Unexpired Insurance Premiums	$ 187.50		
Meeting Expenses	528.60		
		$ 716.10	
			$106,606.50
SPECIFIC FUNDS			
ENDOWMENT FUND			
Cash on Deposit		$13,273.44	
Securities at Cost (Market Value $7,942.25)	$11,918.65		
ADD: Accrued interest	111.35		
		12,030.00	
			25,303.44
F. PAUL ANDERSON AWARD FUND			
Cash on Deposit			1,188.28
RESEARCH FUND			
ENDOWMENT FUND			
Cash on Deposit	$ 517.12		
Cash in Closed Bank	167.78		
		$ 684.90	
UNRESTRICTED FUNDS			
Cash on Deposit	$16,187.55		
Cash on Hand	45.47		
Cash on Hand for Deposit	1,430.16		
		17,663.18	
			18,348.08
			$151,446.30

LIABILITIES AND CAPITAL

SOCIETY			
ACCOUNTS PAYABLE		$23,158.82	
PAYABLE TO RESEARCH FUND			
On Dues as and when Collected		3,796.31	
ACCRUED ACCOUNTS			
Compensation—Secretary and Staff		2,685.43	
DEFERRED INCOME			
Prepaid Dues—Members	$ 549.80		
Prepaid Dues—Proposed Members	273.13		
		822.93	
TOTAL LIABILITIES		$30,463.49	
GENERAL FUND			
Society		76,143.01	
			$106,606.50
SPECIFIC FUNDS			
ENDOWMENT FUND			
Principal		$24,063.28	
Unexpended Income		1,240.16	
			25,303.44
F. PAUL ANDERSON AWARD FUND			
Principal		$ 1,000.00	
Unexpended Income		188.28	1,188.28
RESEARCH FUND			
ENDOWMENT FUND			
Principal	$ 600.00		
Unexpended Income	84.90		
		$ 684.90	
UNRESTRICTED FUND			
Balance—January 1, 1938	$11,606.38		
Excess of Expenses over Income Calendar Year Ended December 31, 1938	6,056.80		
		17,663.18	
			18,348.08
			$151,446.30

NOTE "A"—This Balance Sheet is subject to the comments contained in the letter attached to and forming a part of this report.

BUDGET COMPARISON

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
NEW YORK, N. Y.

For the Calendar Year Ended December 31, 1938

	Actual	*Budget Provision*	*Increases Decreases*
INCOME			
Membership Dues and Initiation Fees	$ 29,106.32	$ 28,000.00	$ 1,106.32
Editorial Contract	15,343.04	14,500.00	843.04
Profit—Emblem and Certificate Frames	98.39	100.00	1.61
Profit—Reprints and Books	368.61	300.00	68.61
Sales of TRANSACTIONS	504.62	500.00	4.62
Interest—Bank	735.79	700.00	35.79
Interest—Securities	305.00	300.00	5.00
GUIDE—Advertising	27,690.66	32,200.00	4,509.34
GUIDE—Copy Sales	26,195.44	30,000.00	3,804.56
	$100,347.87	$106,600.00	$ 6,252.13
EXPENSES			
General Society Activities			
Meetings	$ 1,795.63	$ 2,000.00	$ 204.37
Chapter Allowances	593.24	1,000.00	406.76
Promotional	4,290.70	3,600.00	690.70
Speakers Bureau	1,369.88	2,000.00	630.12
Subscriptions	6,203.85	6,000.00	203.85
Postage	1,850.02	2,500.00	649.98
Rent	3,271.25	4,000.00	728.75
TRANSACTIONS	4,524.42	3,500.00	1,024.42
Codes	218.73	800.00	581.27
Salaries	16,643.00	17,700.00	1,057.00
General Printing	965.12	800.00	165.12
Membership Certificates and Emblems	261.00	600.00	339.00
Travel—President	1,149.25	1,800.00	650.75
Travel—Secretary	1,437.20	1,500.00	62.80
Council and Chapter Delegates Travel	3,711.28	4,500.00	788.72
Bank Charges	74.57	100.00	25.43
Multigraphing	613.45	600.00	13.45
Telephone	754.61	600.00	154.61
Telegraph	418.85	300.00	118.85
Professional Services	400.00	500.00	100.00
General Office Expenses	675.38	600.00	75.38
Office Supplies	356.50	500.00	143.50
Furniture and Fixtures	923.24	1,000.00	76.76
Reserve for Additional Compensation	2,685.43	2,500.00	185.43
Special Appropriation	100.00	1,100.00	1,000.00
	$ 55,286.60	$ 60,100.00	$ 4,813.40
COST OF GUIDE			
Editorial and Advertising Salaries	6,550.00	7,200.00	650.00
Committee Travel	231.36	800.00	568.64
Printing and Binding	15,630.94	20,000.00	4,369.06
Mailing	1,680.00	2,500.00	820.00
Paper	3,483.09	4,500.00	1,016.91
Cuts	436.45	500.00	63.55
Advertising Sales Promotion	1,643.12	2,000.00	356.88
Copy Sales	10,006.70	7,500.00	2,506.70
Yearbook	1,327.22	1,500.00	172.78
	40,988.88	46,500.00	5,511.12
	$ 96,275.48	$106,600.00	$10,324.52
Unbudgeted Appropriations—Research Fund	5,000.00	....	5,000.00
	$101,275.48	$106,600.00	$ 5,324.52

At the conclusion of the report, on motion of Mr. McIntire, seconded by W. L. Fleisher, New York, N. Y., it was voted that the report of the Finance Committee be adopted.

The report of the Membership Committee was persented by R. C. Bolsinger, Philadelphia, Pa., chairman.

Report of Membership Committee

Due to conditions, your committee has not shown as large an increase as desired in membership for the year 1938.

Although membership is the greatest asset of the Society, we have suggested to the various Chapter membership chairmen that numbers is not our chief aim; character and interest are important.

Potential new membership in the South has been pioneered by W. L. Fleisher in the Carolinas and F. E. Giesecke in Texas, which now has two Chapters—the South Texas Chapter at Houston and the North Texas Chapter at Dallas.

We should encourage Pacific Coast activity from British Columbia to lower California by having Council or Society meetings in this section.

It is suggested by this committee that the new Membership Committee give consideration to the idea of appointing zone chairmen to work with the Council Membership Committee so that closer cooperation can be given the outlying Chapter membership chairmen.

During the year 1938, 389 members were elected to the various grades and the present membership status of the Society is as follows:

Honorary Members	2
Presidential Members	21
Life Members	55
Members	1614
Associate Members	780
Junior Members	471
Student Members	59
Total	3002

In closing we want to express our appreciation for the assistance of A. V. Hutchinson and the staff at the headquarters office in carrying out the work of the Committee.

Respectfully submitted,

MEMBERSHIP COMMITTEE

R. C. BOLSINGER, *Chairman.*

President Gurney announced that 1939 marked the 20th anniversary of the establishment of the Society's Research Laboratory and he paid tribute to the work of the devoted men who had carried on this important Society project. He called upon the men present who had served as chairmen of the Committee on Research to arise and be introduced. The following responded: G. W. Barr, Philadelphia; J. I. Lyle, Syracuse; W. H. Driscoll, Syracuse; F. B. Rowley, Minneapolis; C. V. Haynes, Ardmore, Pa.; G. L. Larson, Madison, Wis.; John Howatt, Chicago; A. P. Kratz, Urbana, Ill., and W. A. Danielson, Fort Knox, Ky. Letters and telegrams from those who could not be present were read from Thornton Lewis and F. R. Still, New York; E. E. McNair, St. Petersburg, Fla.; Perry West, Lexington, Ky.; S. R. Lewis, Chicago; and H. P. Gant, Philadelphia.

Mr. Fleisher, chairman of the Committee on Research, was introduced and gave an abstract of the committee's report. At the conclusion of the report, on motion of Mr. Fleisher, seconded by Percy Nicholls, Pittsburgh, Pa., it was moved that the report of the Committee on Research be adopted.

Report of the Committee on Research

Approaching the 20th anniversary of the establishment of the Research Laboratory of the Society at the Pittsburgh Experiment Station of the United States Bureau of Mines in July, 1919, the Committee on Research closes another successful year of service to the heating, ventilating and air conditioning profession and industry. Its activities were administered through 24 Technical Advisory Committees, and while research carried on by some of these groups was limited to Committee deliberation, actual physical research resulting in definite progress was engaged in under a number of others.:

According to plans developed at the Annual Meeting in 1938 the Committee revised its procedure in a number of respects. The Executive Committee was increased from three to five members and took on greater responsibility in the name of the Committee on Research. The Technical Advisory Committees were reorganized and in some instances several Committees reporting parallel interests were grouped under the one heading. As an example, five Committees dealing with definite phases of physiological research, including sensations of comfort for air conditioning in the treatment of diseases and air conditioning requirements of hospitals were combined.

The advantages resulting from this reorganization made it possible for the Chairman of the Committee, with additional assistance, to keep in closer contact with the work of the 24 Technical Advisory Committees, with the result that more consistent and active consideration of the various research projects was possible. The financing of the Committee's work, in line with the general policy of the Society, was simplified in that much of the routine work of accounting was transferred directly under the Treasurer of the Society in the headquarters' office.

An entirely new line of activity dealing with the applications of air conditioning in industry was begun under a new Committee headed by A. E. Stacey, Jr., and was made a major function of the Committee's research activity during 1938. Also the work of establishing fundamentals on which comfort air conditioning requirements are based was actively projected during the current year under C. Tasker's Committee on Sensations of Comfort.

The detailed plans, accomplishments and future objectives of the various Technical Advisory Committees functioning under the Committee on Research during the year 1938 may be referred to in the reports which follow.

Reports of Technical Advisory Committees

SENSATIONS OF COMFORT 1*a*—C. Tasker, *Chairman;* Thomas Chester, F. E. Giesecke, Elliott Harrington, R. E. Keyes, W. J. McConnell, A. B. Newton, J. R. Parsons, C. P. Yaglou.

Two meetings of this Committee were held during 1938. At the request of the Chairman, and with the approval of the Committee, arrangements were made with the Program Committee of the Society to devote a technical session at the Semi-Annual Meeting especially to Comfort Air Conditioning. The papers presented at this meeting are listed elsewhere in this report. Following the interesting results of the Minneapolis study in 1937, arrangements were made to conduct studies in the summer of 1938 in San Antonio, Tex., Washington, D. C., and New York City.

In cooperation with the Texas Agricultural and Mechanical College, the Society sponsored a study in the offices of the San Antonio Public Service Co., San Antonio, Tex. Ninety-four persons (72 men and 22 women) participated in the test, the data being collected by a trained observer thus giving rather better control than that obtainable in earlier studies. The results of this study are being presented at the Annual Meeting in Pittsburgh, January 1939. The results indicate that in spite of mild winters and the intensity and duration of the summer heat in San Antonio there is very little difference in the optimum effective temperature for indoor comfort in Texas as compared with other localities where similar investigations have been made.

In Washington, D. C. the study was conducted at the Federal Reserve Building,

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and the data are now being analyzed by the Research Laboratory. In New York the study was conducted in the Metropolitan Life Insurance Building, New York, under the direction of Dr. W. J. McConnell. The data collected are very comprehensive and the company has volunteered to analyze these data completely. It is hoped that it will be possible to prepare a paper on this study for presentation during the next few months.

In view of the results obtained in the studies made since 1935, it is expected that the next step of the Committee will be to determine the methods of applying the results to actual practice.

PHYSIOLOGICAL REACTIONS 1*b*—C.-E. A. Winslow, *Chairman;* T. Bedford, E. F. DuBois, R. W. Keeton, André Missenard, R. R. Sayers, C. Tasker.

This Committee considers its immediate function to be interpretation to the ventilating engineer of current progress in physiological research, which may influence the solution of his practical problems. During the past year the Committee has prepared a summary of such recent advances as seemed most important, including emphasis on the different reactions of the human body to the environment in various zones of physiological response. In the cold zone, it has stressed particularly the important effects of local chilling of different parts of the body and the very minor influence of relative humidity as compared with air and wall temperatures and air movement. For the hot zone, it has cited the upper limits of temperature and humidity beyond which the evaporative regulatory process of the body is impracticable. It has also discussed problems of comfort and emphasized their intimate relation to physical activity and metabolism. It is the conclusion of the Committee that at present the question of atmospheric adaptation is fundamentally a thermal one, and that there is little ground, under conditions of normal occupancy, for controlling chemical and electrical properties of the atmosphere (ozone or ions). The report of the Committee will be presented at the 1939 Annual Meeting.

In addition to this general summary, Dr. R. W. Keeton and his colleagues have conducted an important experimental study of cardiac output, peripheral blood flow, and blood volume changes in normal individuals subjected to varying environmental temperatures, further results of which will also be presented at the 1939 Annual Meeting.

TREATMENT OF DISEASE 1*c*—T. L. Hazlett, *Chairman;* C. J. Barone, B. Z. Cashman, M. B. Ferderber, E. V. Hill, C. S. Leopold, C. D. Selby, W. O. Sherman, A. W. Sherrill, H. F. Smith, R. J. Tenkonohy, B. L. Vosburgh.

This Committee was organized at the beginning of the year to carry on a study of air conditioning of an operating room and recovery ward in the Magee Hospital in Pittsburgh by the School of Medicine of the University of Pittsburgh in cooperation with the Committee on Research.

Through the cooperation of the interested industry an experimental air conditioning system has been installed to take care of an operating room and recovery ward. The system was put into operation and the collection of data was initiated in June.

The Committee has developed a comprehensive program, including:

1. *Feeling of Warmth:* Optimum temperature and humidity conditions within the operating room and recovery ward for the best interests of the patient, either directly, or through greater efficiency of the operating surgeon and his attendants.

2. *Bacteria:* The bacteria content of the atmosphere in an operating room as found in the past, and with the present air conditioning system; and further, how the bacteria content may be affected by ultra-violet sterilization and electro-static precipitation of dust.

3. *Anesthetic Concentrations:* The relation between past performance without air conditioning and present performance with air conditioning and controlled ventilation, as affecting concentration of anesthetic at different points in the operating room.

4. *Static-Electric Sparks:* Relation between atmospheric conditions and the hazard of electrical sparking possibly causing ignition of air-gas mixtures.

5. *Physiological Aspects of the Subject:* Medical observations relative to weight blood pressures, blood chemistries, and skin temperatures; perspiration of attendants and patients; feeling of comfort, well-being, and presence or absence of fatigue of operators; urine chloride of patient before and after operation; bacteriologic content of operating room air with and without occupants, and effect on healing; and the effect of the recovery room as it aids or deters patient's recovery, which might be contrasted to a room in which no air conditioning exists.

Some data have been collected under several of the items listed, but essentially in connection with item (1), or the determination of the desired effective temperature on the part of doctors, nurses, internes and other attendants, together with the observed reactions of patients to these same conditions. A preliminary paper resulting from the study during the past summer will be presented at the 1939 Annual Meeting.

CLIMATE AND SEASON 1*d*—J. H. Walker, *Chairman;* H. A. Abramson, O. W. Armspach, Ellsworth Huntington, C. A. Mills, André Missenard, T. H. Urdahl, E. L. Weber.

The general objective of this committee is a study of the effect of climate and season on the physiological reactions of human beings, particularly as to their relation to the air conditioning problem. A bibliography has been compiled, which is in the files of the Bureau for Correlating Thermal Research, at the Research Laboratory. Copies may be obtained upon request.

As a contribution to the work of the Committee a paper, dealing with the effect of pollens on health, will be presented at an early meeting of the Society.

AIR CONDITIONING IN INDUSTRY 1*e*—A. E. Stacey, Jr., *Chairman;* Philip Drinker, Leonard Greenburg, H. P. Greenwald, A. M. Kinney, J. W. Kreuttner, L. L. Lewis, W. J. McConnell, C. P. McCord, P. A. McKittrick, R. R. Sayers.

The organization of this Committee was authorized during the current year to study the bearable effective temperatures which a group of people working under prescribed conditions could endure from the standpoint of health. The Committee was also desirous of determining from a sufficiently large group the variations in the physiology of different subjects so that a method of determining those possible divergences in human beings could be established for the selection of a work in a particular industry.

Eleven basic points are enumerated herewith which have an important relationship to proper air conditioning in industry as related to the worker.

1. The simultaneous relationship of temperature and humidity to the reactions of human beings *at work.*
2. The air conditions required for the making of goods, and the air conditions encountered in the extraction of basic materials.
3. The change in the required atmospheric conditions with changes in the manpower output.
4. The increased amount of work healthfully obtained with better conditions.
5. The problem of dust, vapors, fumes and gases which may be harmful to workers.
6. The presence of bacteria, spores, etc., that may produce unsanitary conditions or be harmful to processes.
7. The prevalence of odors arising from working conditions.
8. The prevalence of noise disturbances.
9. Legislation required to protect labor and capital.
10. The engineering basis on which the requirements called for by legislation or determined by research can be calculated.
11. Analysis of research already done or being done or that research required to be done on which to base sound conclusions.

During the past summer the Research Laboratory was assigned to study the physiological reactions of men while carrying on light work in atmospheric conditions, including effective temperature from 77 to 92 deg, with relative humidities of 60, 75 and 90 per cent. It was not only the purpose of this research to determine, if possible, at what point in conditions which may readily occur in industry a physiological break takes place in the workman, but also to outline, from an engineering standpoint, what methods may be pursued, from an air conditioning angle, to relieve or prevent the occurrence of critical conditions. The methods and results produced from this research will be presented in a paper before the 1939 Annual Meeting. Additional work is especially necessary to determine the effect of air movement in reducing the severity of the physiological reactions for different types of work at the same dry-bulb temperature and humidity. In presenting these findings the

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Committee recognizes that the results cover only a few of the factors previously enumerated, which leaves a vast amount of research still to be undertaken for which both local and Government assistance is essential. The Committee also recognizes that industry can assist enormously by offering sections of their plants for laboratory test investigation which would illustrate and confirm or augment the work that has already been done.

AIR CLEANING AND ATMOSPHERIC IMPURITIES 2—H. C. Murphy, *General Chairman.*

MECHANICAL PROCEDURES 2*a*—Leonard Greenburg, *Chairman;* J. J. Bloomfield, W. H. Carrier, R. S. Dill, Theodore Hatch, C. E. Lewis, A. L. Simison, W. O. Vedder.

ELECTRICAL PRECIPITATION AND STERILIZATION 2*b*—E. B. Phelps, *Chairman;* H. E. Adams, R. D. Bennett, L. W. Chubb, L. R. Koller, G. W. Penney, W. F. Wells.

Two papers were presented at the 1938 Semi-Annual Meeting of the Society which cover investigations this Committee is carrying forward. One of these was by R. S. Dill of the U. S. Bureau of Standards and the other was by Prof. F. B. Rowley of the University of Minnesota. A paper outlining further studies on the air filter investigation the Committee on Air Cleaning and Atmospheric Impurities is carrying on at the University of Minnesota will be presented at the 1939 Annual Meeting.

Further studies as to the possibility of revising the ASHVE Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work, to provide an alternate method for rating cleaners by use of the photo-electric cell, are under way. Additional basic research is needed to determine the effects of various dusts and various particle sizes on the absorption of light.

Work on the development of the Code for Testing Air Cleaning Devices Used in Dust Hazardous Occupations has been resumed. This project was deferred temporarily in order that additional data could be secured from the medical profession. It now appears that this work can go forward and a Code be prepared for submission to the Society.

Efforts to coordinate various independent investigations of the control of airborne organisms and the effect of air pollution on epidemic diseases have been instituted.

A meeting of the Committee on Mechanical Procedures was held at the Hot Springs Meeting and also a well attended conference of the entire Committee was held in Washington, D. C., September 30, 1938. At this meeting the Commitee was fortunate in having the active cooperation of the acting chief of the U. S. Public Health Service and his associates. The purpose of the meeting was to consider and further outline a program for research in the field of atmospheric impurities and to endeavor to coordinate this work with other investigating bodies to eliminate useless duplications of effort. The meeting was successful in this respect and it is believed an opportunity for effective cooperation with the various Government bodies has been initiated.

RADIATION AND COMFORT WINTER AND SUMMER AND EFFECT OF VARYING HUMIDITY ON RADIANT HEATING AND COOLING 3—J. C. Fitts, *Chairman;* A. A. Adler, A. H. Barker, W. D. Fleming, R. P. James, C. A. Mills, D. W. Nelson, W. R. Rhoton, W. W. Timmis, G. R. Wait, S. L. Warren, C.-E. A. Winslow, C. F. Wood.

This Committee was appointed the first part of the year to consider all problems in connection with radiation as it affects comfort in winter and summer. As a result of preliminary conferences a long range program of research was outlined to include several basic investigations which are enumerated herewith:

1. A Study based on primary sense reactions of the relative comfort derived from rooms with normal window arrangements from convected heat and from heat supplied by both convection and direct radiation.

2. A study of the effect of radiation in governing the sense of warmth of a person in an occupied space, together with the effect of dry-bulb temperature, moisture content and movement of the air already studied by the Society and included in the Effective Temperature Index or the Comfort Chart.

3. A study of the physiological reactions of persons, including skin temperature, comfort and health measurements, to rooms heated with and without radiation.

4. A study of the relative effect on comfort and physiological reactions by radiation of varying wave-length, and the effect of the relative humidity of the air within the room on the effectiveness of such radiation.

As an immediate objective the Committee has been considering plans for the initiation of studies covering item 1 to be conducted at the Research Laboratory in Pittsburgh.

WEATHER DESIGN CONDITIONS 5—T. H. Urdahl, *Chairman;* J. C. Albright, F. S. Cornell, John Everetts, Jr., E. W. Goodwin, A. C. Grant, J. B. Kincer, O. A. Kinzer, J. W. O'Neill, Leon Ourusoff.

This Committee has made much greater progress during the current year than its limited resources seemed to permit at the time the program was initially adopted. This progress is due largely to having had made available a great quantity of summer design temperature data which had been previously analyzed and the cooperative continuance of the work to embrace the analysis of some additional data required in attaining the Committee's objectives.

A detailed presentation of the accomplishments of the Committee will be presented in a paper at the Annual Meeting in Pittsburgh, January 1939, and the following indicates results along the line of procedure set up in the Committee's original program.

1. A thorough study and analysis of available basic data from the records of the U. S. Weather Bureau and other sources has indicated to what extent existing data are adequate for the purpose of heating and air conditioning design; and also, the causes of error in existing design weather data. It will *now* be possible to prepare a brief bibliography of available data.

2. The factors which apparently were not given full consideration in previous studies have been brought out in the course of analysis and should be given further consideration in future work. These are chiefly:

A. Wind velocities at lowest design dry-bulb temperatures in winter, and effect of solar radiance.

B. The possibility, duration, and effect of coincidental high dry-bulb and wet-bulb temperatures and wind velocities in summer.

C. Temperature differences between business and suburban districts, due to higher temperature conditions resulting in the former through trapping of solar heat in buildings and pavements, etc., and the difference between these temperatures and those taken at the local weather bureau station by duration of extremes and the time of their occurrence; i.e., day, night, morning or afternoon.

3. The work done under the auspices of the Committee in analyzing almost 3,000,000 temperature readings has resulted in the adoption of methods of collecting and analyzing weather data from which uniform basis of future collection and analysis may be recommended.

4. Methods of obtaining data were considered and used as follows:

A. The U. S. Weather Bureau at Washington, D. C., made available to the Committee all of its records and also generously permitted the use of space in its office to employees of the Committee in abstracting temperature data from original records.

B. The Utilities, with which there has been some discussion of the Committee's program, indicate a willingness to cooperate in carrying out the future program which is required to obtain data not now available.

The results of the Committee's work at the present time include, besides the procedures outlined in the foregoing, certain valuable additions to existing design weather data which will be available next year. These data include not only corrections of existing design temperatures, but also quantitative data as to the number of hours and percentage of time during summer months in which various wet-bulb, dry-bulb, and dew-point temperatures occur, also wind velocities and hours of sunshine.

Possibly the most valued objective gained through the work of this Committee, to date, is the establishment of a basis upon which to proceed with future work that is required. While it is immediately apparent that an immense amount of time has been spent in making these data available, it is nevertheless true that this work represents only a beginning of the ultimate project. The objectives of the future program of this Committee are as follows:

1. To establish uniform methods of obtaining and analyzing weather data upon which design conditions for any given locality may be based.

2. To quantitatively analyze existing weather data including dry- and wet-bulb temperatures, wind velocity, and sunshine, in a manner and over a sufficient period of years to show range and duration by months and seasons following the procedure already employed in obtaining some of the summer data.

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3. To supplement the work being done by the U. S. Weather Bureau through obtaining those data which they do not have available. The chief deficiency is wet-bulb data which are only available for approximately four years in most localities. This supplemental work to consist of the taking of wet- and dry-bulb temperatures at a central point in each locality at the street level and at the average roof level for comparison with local U. S. Weather Bureau data, and in order also to determine any existing temperature differences between business and suburban sections and the influence of solar heat storage in massed pavements, buildings, etc.

4. To make available the information obtained in the foregoing to each locality through assembly, analysis, and distribution from one central point until such time as such work may be done by the Government as a part of its weather reporting program.

5. To recommend to the U. S. Government desired additions to and changes in the data being obtained through the weather bureau stations, and to support the required legislative effort as may be necessary to expand the activity of this bureau to include the reporting and dissemination of the data required in order to form a proper and adequate basis for design of heating and air conditioning systems.

TRANSPORTATION AIR CONDITIONING 6—J. H. Van Alsburg, *Chairman;* W. I. Cantley, T. R. Crowder, A. G. Dixon, C. C. Elmes, L. H. Laffoley, E. A. Russell, W. E. Zieber.

This Committee was continued for the year 1938 with the hope that active cooperation could be obtained from various railroad organizations to conduct extensive research work in this field. However, due to the inability of the Committee to obtain assistance from the railroad interests it was found inadvisable to initiate any studies this year, and consideration has been given to reviewing reports which have been previously published covering railroad air conditioning and preparing a report of recommendations for future research studies. The Committee has also given some consideration to preparing a research program on bus ventilation in the District of Columbia under the supervision of one of the Federal Agencies.

RADIATION WITH GRAVITY AIR CIRCULATION 7—M. K. Fahnestock, *Chairman;* B. C. Benson, H. F. Hutzel, J. P. Magos, J. W. McElgin, J. F. McIntire, D. W. Nelson, T. A. Novotney, W. A. Rowe.

The work under this committee during the past year has been a continuation of a program which has been in progress for a number of years. The program may be divided into two parts: *First,* that which is being conducted in the warm wall test booth, and *Secondly,* that which may be conducted in the room heating testing plant.

Under the former the study of the size of the inlet and outlets of convector heater cabinets with reference to their effect upon the heat output or capacity of the units was completed, and the data are now being analyzed and prepared for presentation in a paper before the Society. This study was made with three distinct types of heating elements, and some twenty different combinations of inlets and outlets were tested in the cabinet used on each of the three heating elements. It is hoped that the results will form the basis for designing convector cabinets having proper inlets and outlets for maximum heat output.

Under the second study the remodeling of the room heating testing plant has been practically completed and it will be placed in operation as soon as the calibration of the numerous thermocouples has been made. A thermal integrator and a globe thermometer are being constructed to determine the equivalent temperature of the environment when the test room is being heated by various types of radiators and convectors. These instruments will also be used in studying the radiation present in the environment.

The immediate program to be carried out in the test plant includes:

1. Tests to correlate the performance of conventional tubular steam radiators (sections on 2½-in. centers) in the new 15 ft x 18 ft x 8 ft 6 in. test room with their performance in the 9 ft x 11 ft x 9 ft test rooms which were in the old plant.

2. Tests to determine the performance of thin radiators free standing in the room under windows in exposed walls, and in recesses under windows in exposed walls.

3. Tests to determine the performance of several of the new light-weight or Junior size (sections on less than 2½-in. centers) of direct steam radiators, including their capacities and their distribution of heat in the room.

Since the room heating testing plant is now completed it is hoped that members of the committee, and the industry, will submit new problems adaptable to the facilities afforded by it.

HEAT TRANSFER OF FINNED TUBES WITH FORCED AIR CIRCULATION 8—G. L. Tuve, *Chairman;* W. E. Heibel, H. F. Hutzel, R. F. Norris, R. H. Norris, C. H. Randolph, L. P. Saunders, C. F. Wood.

Work under this Committee was continued at Case School of Applied Science and a paper was presented at the 1938 Semi-Annual Meeting. In this paper methods were reviewed which are available at the present time for selecting dehumidifying coils, and a new procedure was outlined which is referred to in the paper as the *humidity method.*

Further studies are being continued to determine the effect of refrigerant temperature on the performance of dehumidifying coils. Five sub-divisions of the general problem are covered in the present plan for experimental studies and correlation.

1. Variation of air-side surface coefficients of both dry and wet coils.
2. Effect of water velocities on the water-side coefficient of both refrigerating and heating coils, connected in counter-flow and in cross-flow.
3. Improvement of methods for measuring refrigerant-side surface coefficients of direct-expansion coils.
4. Effect of coil depth or number of rows of tubes, and performance of very deep coils (6 to 10 rows).
5. Relation of heat transfer to air friction and air velocity.

The entire program is being directed towards the formulation of uniform practices which eventually can be used as a basis for a standard Society code for rating and selecting dehumidifying coils.

COOLING LOAD IN SUMMER AIR CONDITIONING 9—J. H. Walker, *Chairman;* C. M. Ashley, John Everetts, Jr., F. H. Faust, A. E. Knapp, L. S. Morse, A. E. Stacey, Jr., R. M. Stikeleather.

This committee has been interested in the effect of heat transfer through walls and windows resulting from solar radiation, on the cooling load in summer air conditioning. So far, the committee has been interested in a few preliminary tests to determine the type of data needed and the analysis of results from various sources. From this work it has developed a comprehensive program for determining the needed information and methods of reporting it for design purposes. The work will be initiated as soon as the necessary budget is provided.

SOLID FUELS 10—W. A. Danielson, *Chairman;* H. N. Eavenson, A. C. Fieldner, A. J. Johnson, P. Nicholls, V. F. Parry, H. J. Rose, J. E. Schoen, L. E. Seeley, E. T. Selig, R. A. Sherman, R. Templeton Smith, C. Tasker.

This Committee was organized shortly after the 1938 Annual Meeting in New York. It has been thought for some time that the Society should take a more active interest in fuels from the standpoint of the user. Coal and its solid products, and to a lesser extent wood, make up the major portion of the fuels used throughout the world and for that reason the Committee on Research recommended the initiation of a study on that class of fuels. A comprehensive outline of the program that is now being studied has been prepared. Arrangements were made for a technical session at the 1939 Annual Meeting to be devoted to this subject.

Two sub-committees have been formed, one on anthracite headed by A. J. Johnson, and the other on bituminous coal and coke under the direction of R. A. Sherman. These Committees will determine methods of comparing performance of fuels and equipment under the various items of *consumer satisfaction,* will develop and propose minimum performance specifications, and will cooperate with other research agencies. A sub-committee on bibliography consists of H. J. Rose, A. C.

Fieldner and R. A. Sherman. This group has made plans to prepare short annotated bibliographies of recent articles on selected subjects relating to solid fuels and their utilization. Another sub-committee on Chimneys consists of the following personnel: A. J. Johnson, chairman, J. H. Billings, L. E. Moody, Stanley Morehouse, H. C. Plumber and G. B. Thom.

SUMMER AIR CONDITIONING FOR RESIDENCES 11—M. K. Fahnestock, *Chairman;* E. A. Brandt, John Everetts, Jr., Elliott Harrington, H. F. Hutzel, E. D. Milener, K. W. Miller, R. E. Robillard, F. G. Sedgwick, J. H. Walker.

The research work of this committee has been in progress continuously for the past seven years, and the results, not including those obtained during 1938, have appeared in seven technical papers presented before the Society and published in its JOURNAL and TRANSACTIONS. An eighth paper covering the work done during the summer of 1938 is being prepared. It covers the following items, some of which are a continuation of work started in 1937.

1. Tests to obtain additional information on the practicability of using a small capacity mechanical condensing unit in warm weather. Unfortunately, the cooling season of 1938, like that of 1937, did not afford any weather on which the outdoor temperature exceeded 95 F. However, by starting the cooling unit earlier in the day, when the indoor temperature reached 77 F, more desirable indoor conditions were maintained than in 1937.

2. Two series of tests were made for the purpose of determining the conditions resulting on both the first and second stories, when the conditioned air was all introduced on the second story. In one series of tests the cooling capacity of the plant was 18,000 Btu per hour and in the other it was 30,000 Btu per hour. The conditions maintained in the latter series of tests are to be compared with those attained in 1934 when the conditioned air from a plant having a capacity of 30,000 Btu per hour was introduced on both stories.

3. Using a cooling capacity of 30,000 Btu per hour tests were conducted to obtain data on the rate at which the temperature and humidity could be reduced, after they had attained values materially above those conducive to comfort.

4. Using a Nicholls' heat-flow meter, data were obtained on the amount of heat passing through a section of the frame wall during entire 24-hour periods, where the studding space was filled with rock wool. These data are very interesting when compared with those obtained in 1937 on the same wall section with the studding spaces empty.

AIR DISTRIBUTION AND AIR FRICTION 12—J. H. Van Alsburg, *Chairman;* S. H. Downs, M. K. Fahnestock, F. J. Kurth, R. D. Madison, L. G. Miller, D. W. Nelson, C. H. Randolph, Ernest Szekely, G. L. Tuve.

Results from continued research extending over a period of three years are being presented by the Laboratory in a paper at the 1939 Annual Meeting on frictional resistance to the flow of air in straight ducts. This paper presents two friction charts, one of which may be used for future research investigations, and the other for practical application work in the field. The studies on duct friction are not complete at the present time as further work is planned for investigations on rectangular ducts, elbows and numerous other fittings.

A comprehensive program for investigations in the field of air distribution has been planned for initiation at various cooperative institutions and the Laboratory. At the Case School of Applied Science a small wind tunnel has been equipped with a Thomas meter, orifice and Pitot tube so tests can be conducted at very low velocities to determine instruments most suitable for recording low air velocities. At this institution studies are also being made on the behavior of air discharged from grilles into large and small unconfined spaces.

A project has been outlined at the University of Wisconsin to determine the most suitable types of fittings which may be used to produce equal air flow over the face of an opening, and also to determine the resistance characteristics for the several types of fittings. At the University of Illinois it has been planned that tests will be conducted in a cold room where the walls, ceilings and floors may be heated or cooled in order to determine the advantages of location of air outlet with specific regard to supply and return.

HEAT REQUIREMENTS OF BUILDINGS 13—P. D. Close, *Chairman;* W. H. Badgett, E. K. Campbell, E. F. Dawson, W. H. Driscoll, H. M. Hart, H. H. Mather, F. B. Rowley, R. J. J. Tennant, J. H. Walker.

The activities of this Committee for the current year have included projects dealing with intermittent heating, heat losses into the ground, selection of temperatures and wind velocities, and heat losses through top floor ceilings under pitched roofs.

Arrangements have been made for a study on intermittent heating to be conducted at the Agricultural and Mechanical College of Texas. Guion Hall, the College auditorium, is being used for this purpose and is heated by hot water from the central heating plant. A hot water flow meter has been installed as well as two pen recording thermometers in the flow and return mains to determine the quantity of heat supplied to the building. Recording and indicating thermometers have been placed throughout the building to measure air temperatures, and thermocouples have been placed in the exterior and interior walls, floor, ceiling and roof to determine temperature gradients through them. During periods of low outdoor temperature the building, with heat off, is opened and allowed to cool to as low a uniform temperature as possible; then closed and reheated to a uniform temperature. From these and other observed data, an effort is being made to determine the number of Btu per square foot of surface per degree temperature difference (indoor to outdoor) of the walls, floor and interior partitions for the construction of the building, and then for other typical building constructions, required to heat them to certain required temperatures in a given time. These results will be checked against theoretical calculations.

In connection with heat losses into the ground more reliable data are needed on the calculation of heat losses through basement floors into the ground. A study of this subject at the Research Laboratory is under consideration.

Present methods of calculating heat losses involve the selection of temperature and wind velocities which may have no relationship to each other and may not necessarily be the combination which will result in the maximum heat requirement. It is the opinion of some engineers that concurring combinations of temperature and wind velocity should be selected for computing heat losses, and such concurring combinations should be based on actual weather bureau records for each locality. An analytical study of this character for important cities in the United States is under consideration.

It is the practice in many cases to estimate the heat loss through top floor ceilings by assuming the attic temperature to be the mean between the inside and outside temperatures. In the case of attics with thick insulations between the ceiling joists, the attic temperature will ordinarily be only a few degrees above the outside temperatures rather than the mean between the inside and outside temperatures. Therefore, calculations based on the latter assumption are likely to be somewhat in error. A formula for calculating attic temperatures is proposed:

$$t_a = \frac{A_c U_c t_i + (A_r U_r + A_w U_w + A_g U_g) t_o}{A_r U_r + A_w U_w + A_g U_g + A_c U_c}$$

where:

t_a = attic temperature, degrees Fahrenheit.
t_i = inside temperature near top floor ceiling, degrees Fahrenheit.
t_o = outside temperature, degreess Fahrenheit.
A_c = Area of ceiling, square feet.
A_r = Area of roof, square feet.
A_w = Area of net vertical wall surface, square feet.
A_g = Area of glass, square feet.
U_c = Coefficient of transmission of ceiling, based on surface coefficient of 2.20 (upper surface).
U_r = Coefficient of transmission of roof, based on surface coefficient of 2.20 (lower surface).
U_w = Coefficient of transmission of vertical wall surface.
U_g = Coefficient of transmission of glass.

Example: Calculate the temperature in an unheated attic, assuming the following conditions: $t_i = 70$; $t_o = 10$; $A_c = 1000$; $A_r = 1200$; $A_w = 100$; $A_g = 10$; $U_r = 0.50$; $U_c = 0.40$; $U_w = 0.30$; $U_g = 1.13$.

Solution: Substituting these values in the previous formula:

$$t_a = \frac{(1000 \times 0.40 \times 70) + 10[(1200 \times 0.50) + (100 \times 0.30) + (10 \times 1.13)]}{(1200 \times 0.50) + (100 \times 0.30) + (10 \times 1.13) + (1000 \times 0.40)}$$

$$t_a = \frac{28{,}641}{1041} = 27.4 \text{ F.}$$

This formula does allow for attic infiltration, but this factor could readily be provided for, if deemed advisable, by introducing another term.

AIR CONDITIONING REQUIREMENTS OF GLASS 14—M. L. Carr, *Chairman;* F. L. Bishop, A. N. Finn, S. O. Hall, E. H. Hobbie, R. A. Miller, F. W. Parkinson, Harold Perrine, W. C. Randall, L. T. Sherwood, J. T. Staples, C. Tasker, G. B. Watkins, F. C. Weinert.

This Committee has been very active during the past year and has held two meetings, one at the time of the Annual Meeting and one at the time of the Semi-Annual Meeting. The Committee has maintained its own budget and has employed a research assistant at the Society Laboratory.

To date, no independent research has been carried on, but instead, the Committee has been interested in analyzing the results of research carried on in various branches of the industry. In this connection, two papers have been released for publication which are referred to in this report. The Committee has also developed a test specification for determining heat transfer through windows and glass block structures.

INSULATION 15—W. A. Danielson, *Chairman;* E. A. Allcut, R. E. Backstrom, H. C. Bates, Wharton Clay, H. C. Dickinson, J. D. Edwards, W. V. Hukill, E. C. Lloyd, Paul McDermott, E. W. McMullen, R. T. Miller, W. T. Miller, E. R. Queer, T. S. Rodgers, F. B. Rowley, W. S. Steele, R. J. Tenkonohy, G. B. Wilkes.

This Committee has considerable research work under investigation in a number of laboratories. The most extensive of these is at the University of Minnesota under the guidance of Prof. F. B. Rowley. The importance of the water vapor barrier on the warm side of an insulated enclosed space, and even in so-called uninsulated space where the humidity in the warm space is appreciably above that on the cooler side, has been definitely demonstrated. At the several meetings of the Committee, this situation was discussed at length and is at present the most important phase in the use of insulation. The Committee has also reviewed carefully the portion of THE GUIDE devoted to heat transmission and has made pertinent recommendations to the Guide Committee. This part of the work is being continued through a sub-committee. Basic studies on the relationship of structures to insulating values and evacuated spaces are being made under a cooperative agreement with the University of Toronto.

SOUND CONTROL 16—J. S. Parkinson, *Chairman;* C. M. Ashley, G. F. Drake, A. M. Greene, Jr., A. L. Kimball, V. O. Knudsen, R. F. Norris, C. H. Randolph, J. P. Reis, W. P. Roop, A. E. Stacey, Jr., G. T. Stanton, F. R. Watson.

This Committee has devoted the larger part of its energies during the past year to the establishment of a test code for rating ventilating and air conditioning apparatus in terms of the noise generated by such apparatus. The object of such a code would be to provide a numerical decibel rating for each type of equipment which would (*a*) enable the prospective user or purchaser to compute the probable amount of noise which would be generated in a given location by this apparatus, and (*b*) would enable such a purchaser to compare the merit of any given piece of equipment with other competitive apparatus. The establishment of such a code involves as a corollary the development of a testing technique.

A preliminary code was drawn up and submitted to the members of the Committee for their comments. When these had been received, the revised code was drafted and this has been resubmitted to the Committee members. The first study contemplated in the research on the acoustical problems, being conducted at Princeton University, will be an investigation of the testing techniques proposed in the apparatus test code.

The Committee has also assembled such information as was available on the acoustical behavior of ventilating and air conditioning equipment. As part of its activities, it undertook a review of the chapter on Sound Control in The GUIDE 1939.

COOLING TOWERS, EVAPORATIVE CONDENSERS AND SPRAY PONDS 17—B. M. Woods, *Chairman;* J. C. Albright, S. C. Coey, E. R. Goodrich, E. H. Hyde, E. H. Kendall, S. R. Lewis, O. W. Ott, E. H. Taze.

Through a cooperative agreement with the Society investigations have been initiated at the University of California, in Berkeley, to study four types of forced draft and atmospheric type cooling towers. Two atmospheric towers have been placed on the roof of Hesse Hall on a rotating platform. The roof is shielded from wind by means of louvres. One of the forced convection towers employs *sheet* cooling and the other *drop* cooling.

High precision instruments have been developed and constructed for this project, and flow orifices for both air and water have been installed to record the results. Instruments which have been designed for this project include a portable hair hygrometer, a thermocouple hygrometer and a radiation meter. A pressure plate which possesses a high natural frequency and which will be used to record wind velocity variations is being designed for the study. An Assman psychrometer and a Tagliabue potentiometer are available for the project.

A technical report for publication by the Society is in preparation based on a complete set of performance test data which are now available on one of the towers. The nature of the packing in this tower is such as to cause the cooling of the water at the surface of water sheets (not drops). An analytical prediction of this tower yielded results which were in satisfactory agreement with the test data.

Data are being collected to establish a minimum rate of evaporation which will occur from a stationary, horizontal or vertical water surface into stationary air. Computations which will establish the rate of fall for water drops of different diameters are being made. From these results cooling rates may be predicted in any tower provided the drop size distribution is known. The drop size distribution in a tower will be obtained with the aid of high speed photography. These results should be of direct aid for tower design.

The Committee recognizes the importance of a study of evaporative condensers. It plans a review of the literature and status of design during the coming year from which an investigational program with laboratory experiments can be formulated. One member of the Committee is conducting experiments on water cooled roofs and has submitted a preliminary report. Work will be continued in the summer of 1939.

PSYCHROMETRY 18—F. R. Bichowsky, *Chairman;* C. A. Bulkeley, J. A. Goff, A. M. Greene, Jr., F. G. Keyes, D. W. Nelson, W. M. Sawdon.

The general objective of this work is to prepare a thermodynamically consistent table of the properties of air and such charts based on this table as appear to be most useful. Dalton's Law of Mixtures has been universally employed in the properties of steam-air mixtures. A thermodynamic study of the properties of steam-air mixtures indicates that this assumption is not justifiable under the range of mixtures covered by the psychrometric chart and that therefore the standard tables for the properties of steam-air mixtures are seriously in error, the errors amounting to several degrees wet-bulb in some cases. For this reason the committee has undertaken an experimental determination of the deviations of steam-air mixtures from Dalton's Law, and has under consideration an experimental program covering development

of a new type of wet-bulb instrument, calibration of this instrument, and direct determination of the heat of vaporization of water into air streams.

CORROSION IN STEAM SYSTEMS 19—A. R. Mumford, *Chairman;* H. E. Adams, J. F. Barkley, W. H. Driscoll, T. J. Finnegan, R. M. Palmer, R. R. Seeber, F. N. Speller, C. M. Sterne.

This group has carefully reviewed its program and decided to specialize for this year on one phase of the work.

The single phase selected for closer study is the determination of the amount of nine-condensible gases which will dissolve in the condensate formed in a heating system at different rates of condensation when the steam contains a known amount of these non-condensible gases. This work is being conducted at the Michigan College of Mining and Technology. Another phase of the work which has been undertaken is a study of the operating cycle of typical hot-water heaters. This cycle includes a study of the variations in the rate of steam and water flow, condensate temperatures and related information. A progress report on the study of one such heater is expected to be ready for publication in the near future.

Interested manufacturers and steam-producing companies have been informed of a proposed project to determine the rate of corrosion caused by condensate formed in a partial pressure of carbon dioxide in the absence of oxygen. This project was proposed in order to determine whether the solution of iron by fresh condensate would buffer it sufficiently to render it non-corrosive shortly after it entered the return line system. It may not be possible to undertake this program at the present time, but later developments may change this preliminary conclusion because of the interest expressed by certain companies.

CORROSION IN AIR CONDITIONING EQUIPMENT 20—A. E. Stacey, Jr., *Chairman;* A. F. L. Anderson, G. L. Cox, M. L. Diver, W. R. Heath, C. E. Lewis, R. M. Palmer, F. N. Speller, C. M. Sterne, J. H. Young.

At a meeting held in January 1938, it was decided that the best approach to studying corrosion in air conditioning equipment would be to locate samples of different materials with various protective coatings, both non-metallic and metallic, in apparatus located in different parts of the country As these test samples had to be prepared by different individuals in several localities, the test samples were not available for testing during the 1938 operating season. It is the plan of the Committee to insert these samples in various equipment during the spring of 1939 in order that comparative results may be available for publication the latter part of the year.

The Committee has cooperated with 17 other technical societies in conferences intended to coordinate all of the corrosion activities in this country. Through the efforts of this group it is expected that all of the organizations actively engaged in corrosion may coordinate their efforts and thus avoid duplication.

RESEARCH FUND

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
NEW YORK, N. Y.

For the Calendar Year Ended December 31, 1938

(From Report of Certified Public Accountants, Tusa & LaBella, New York, N. Y.)

STATEMENT OF INCOME AND EXPENSES

INCOME

CONTRIBUTIONS		
From General Fund—40% of Dues Collected	$16,955.88	
From Outside Sources—Per Schedule	4,885.00	$21,840.88
APPROPRIATIONS		
From General Fund		5,000.00
1938 EXPOSITION		9,673.56
INTEREST ON BANK DEPOSITS		60.91
		$36,575.35

EXPENSES

AT PITTSBURGH, PA.		
Salaries—Laboratory Staff	$12,969.50	
Salaries—Graduate Student Help	1,893.73	
Salaries—Office of Chairman	11.25	
Correlating Thermal Research	1,200.00	
Laboratory Supplies and Equipment	697.92	
Traveling	2,254.60	
Publicity and Promotion	692.50	
Meetings	289.97	
Office Supplies and Expenses	539.52	
Exhibits	752.54	
Printing	229.72	$21,531.25
COOPERATIVE RESEARCH		
General		
A. & M. College of Texas	$ 950.00	
Case School of Applied Science	700.00	
Princeton University	250.00	
University of California	600.00	
University of Illinois	600.00	
University of Minnesota	625.00	
University of Pittsburgh	375.00	
University of Toronto	500.00	
University of Wisconsin	300.00	
Weather Design Committee	300.00	
Medical		
University of Illinois	1,575.00	6,775.00
OTHER RESEARCH		
Glass Study		2,012.30
OTHER EXPENSES		
Weather Design Committee		200.00 $30,518.55
EXCESS OF INCOME OVER EXPENSES		$ 6,056.80

BUDGET COMPARISON

	Actual	*Budget Provision*	*Increases Decreases* *
INCOME			
Balance from 1937 Committee—Cash in Pittsburgh and New York Banks	$11,606.38	$11,606.38	....
Dues—Current and Prior Years	16,955.88	15,000.00	$ 1,955.88
Special Appropriations	5,000.00	5,000.00	....
1938 Exposition	9,673.56	3,500.00	6,173.56
Interest on Savings Account	60.91	....	60.91
Contributions—General Research	1,675.00	....	1,675.00
Contributions—Air Distribution	1,600.00	....	1,600.00
Contributions—Corrosion Steam Systems	550.00	....	550.00
Contributions—Glass Study	1,010.00	....	1,010.00
Contributions—Intermittent Heating	50.00	....	50.00
	$48,181.73	$35,106.38	$13,075.35
EXPENSES			
Travel—Committee and Laboratory Staff	$ 2,254.60	$ 2,000.00	$ 254.60
Salaries—Laboratory Staff	12,969.50	13,500.00	530.50*
Salaries—Student Help	1,893.73	1,800.00	93.73
Salaries—Office of Chairman	11.25	300.00	288.75*
Cooperative Research—General	5,200.00	5,475.00	275.00*
Cooperative Research—Medical	1,575.00	2,100.00	525.00*
Laboratory Supplies and Equipment	697.92	700.00	2.08*
Office Supplies and Equipment	539.52	500.00	39.52
Printing	229.72	250.00	20.28*
Exhibits	752.54	750.00	2.54
Correlating Thermal Research	1,200.00	1,200.00	....
Promotion and Publicity	692.50	1,000.00	307.50*
Meetings	289.97	250.00	39.97
Contingencies	200.00	500.00	300.00*
	$28,506.25	$30,325.00	$ 1,818.75*
Earmarked for Glass Study	2,012.30	3,411.93	1,399.63*
	$30,518.55	$33,736.93	$ 3,218.38*
Balance	17,663.18	1,369.45	16,293.73
	$48,181.73	$35,106.38	$13,075.35

Papers Resulting from Research During 1938

The 24 research projects considered by as many Technical Advisory Committees resulted in 21 papers during the year, as listed below. The research from which these papers resulted was carried out at the Research Laboratory in Pittsburgh, by the Laboratory in cooperation with other organizations and by Universities in cooperation with the Committee on Research. These papers, resulting from the Society's research activities, are becoming more and more an important part in the deliberations of the Society meetings, and are looked forward to with greater expectation and confidence by the membership of the Society as being vitally important in the progress of this industry.

1. Draft Temperatures and Velocities in Relation to Skin Temperature and Feeling of Warmth. by F. C. Houghten, Carl Gutberlet and Edward Witkowski. RESEARCH REPORT No. 1086, ASHVE TRANSACTIONS, Vol. 44, 1938.

2. Physiologic Response of Man to Environmental Temperature, by F. K. Hick, R. W. Keeton and Nathaniel Glickman. RESEARCH REPORT No. 1079, ASHVE TRANSACTIONS, Vol. 44, 1938. (Results of cooperative research at College of Medicine, University of Illinois.)

3. Air Distribution from Side Wall Outlets, by D. W. Nelson and D. J. Stewart. RESEARCH REPORT No. 1076, ASHVE TRANSACTIONS, Vol. 44, 1938. (Results of cooperative research at the University of Wisconsin.)

4. Study of Summer Cooling in the Research Residence Using a Small Capacity Mechanical Condensing Unit, by A. P. Kratz, S. Konzo, M. K. Fahnestock and E. L. Broderick. RESEARCH REPORT No. 1095, ASHVE TRANSACTIONS, Vol. 44, 1938. (Results of cooperative research at the University of Illinois.)

5. Cooling Requirements for Summer Comfort Air Conditioning in Toronto, by C. Tasker. RESEARCH REPORT No. 1100, ASHVE TRANSACTIONS, Vol. 44, 1938. (Results of cooperative research at the Ontario Research Foundation.)

6. Heat Transfer Through Single and Double Glazing, by M. L. Carr, R. A. Miller, Leighton Orr and Alan C. Byers. RESEARCH REPORT No. 1096, ASHVE TRANSACTIONS, Vol. 44, 1938.

7. Shock Experiences of 275 Workers After Entering and Leaving Cooled and Air Conditioned Offices, by A. B. Newton, F. C. Houghten, Carl Gutberlet, R. W. Qualley and M. C. W. Tomlinson. RESEARCH REPORT No. 1102, ASHVE TRANSACTIONS, Vol. 44, 1938.

8. Air Filter Performance as Affected by Kind of Dust, Rate of Dust Feed and Air Velocity Through Filter, by Frank B. Rowley and Richard C. Jordan. RESEARCH REPORT No. 1094, ASHVE TRANSACTIONS, Vol. 44, 1938. (Results of cooperative research at the University of Minnesota.)

9. General Reactions of 274 Office Workers to Summer Cooling and Air Conditioning, by F. C. Houghten, A. B. Newton, R. W. Qualley and Edward Witkowski. RESEARCH REPORT No. 1103, ASHVE TRANSACTIONS, Vol. 44, 1938.

10. Performance of Surface-Coil Dehumidifiers for Comfort Air Conditioning, by G. L. Tuve and L. J. Seigel. RESEARCH REPORT No. 1099, ASHVE TRANSACTIONS, Vol. 44, 1938. (Results of cooperative research at Case School of Applied Science.)

11. The Flow of Air Through Exhaust Grilles, by A. M. Greene, Jr. and M. H. Dean. RESEARCH REPORT No. 1092, ASHVE TRANSACTIONS, Vol. 44, 1938. (Results of cooperative research at Princeton University.)

12. Seasonal Variations in Effective Temperature Requirements, by F. E. Giesecke and W. H. Badgett. RESEARCH REPORT No. 1101, ASHVE TRANSACTIONS, Vol. 44, 1938. (Results of cooperative research at Agricultural and Mechanical College of Texas.)

13. A Study of the Heat Requirements of a Single-Glazed Test House and a Double-Glazed Test House, by M. L. Carr, R. A. Miller, Leighton Orr and David Shore. RESEARCH REPORT No. 1113, ASHVE TRANSACTIONS, Vol. 45, 1939.

14. Frictional Resistance to the Flow of Air in Straight Ducts, by F. C. Houghten, J. B. Schmieler, J. A. Zalovcik and N. Ivanovic. RESEARCH REPORT No. 1105, ASHVE TRANSACTIONS, Vol. 45, 1939.

15. Air Conditioning in Industry, by W. L. Fleisher, A. E. Stacey, Jr., F. C. Houghten and M. B. Ferderber. RESEARCH REPORT No. 1106, ASHVE TRANSACTIONS, Vol. 45, 1939.

16. Cardiac Output, Peripheral Blood Flow, and Blood Volume Changes in Normal Individuals Subjected to Varying Environmental Temperatures, by F. K. Hick, R. W. Keeton, Nathaniel Glickman and H. C. Wall. RESEARCH REPORT No. 1108, ASHVE TRANSACTIONS, Vol. 45, 1939. (Results of cooperative research at the College of Medicine, University of Illinois.)

17. Air Conditioning Requirements of an Operating Room and Recovery Ward, by F. C. Houghten and Dr. W. Leigh Cook. RESEARCH REPORT No. 1111, ASHVE TRANSACTIONS, Vol. 45, 1939. (Results of cooperative research at the University of Pittsburgh.)

18. Heat Transfer in Storage Water Heaters, by D. W. Nelson and A. A. Rosenberg. RESEARCH REPORT No. 1114, ASHVE TRANSACTIONS, Vol. 45, 1939. (Results of cooperative research at the University of Wisconsin.)

19. Air Filter Performance as Affected by Low Rate of Dust Feed, Various Types of Carbon, and Dust Particle Size and Density, by F. B. Rowley and R. C. Jordan. RESEARCH REPORT No. 1122, ASHVE TRANSACTIONS, Vol. 45, 1939. (Results of cooperative research at the University of Minnesota.)

20. Reactions of Office Workers to Air Conditioning in South Texas, by A. J. Rummel, F. E. Giesecke, W. H. Badgett and A. T. Moses. RESEARCH REPORT No. 1127, ASHVE TRANSACTIONS, Vol. 45, 1939. (Results of cooperative research at the Agricultural and Mechanical College of Texas.)

INSTITUTIONS COOPERATING WITH THE COMMITTEE ON RESEARCH

Agricultural & Mechanical College of Texas

Summer Comfort Cooling Requirements.
Heating Requirements of Buildings.

Case School of Applied Science

Heat Transfer of Finned Tubes with Forced Air Circulation.
Air Distribution in Air Conditioned Spaces.

University of California

Performance of Cooling Towers.

University of Illinois

Air Conditioning in the Treatment of Diseases.
Direct and Indirect Radiation with Gravity Air Circulation.
Residence Air Conditioning.
Air Distribution Outlets.

Michigan College of Mining & Technology

Corrosion in Steam Heating Systems.

University of Minnesota

Air Cleaning Devices.
Conductivity of Insulation and Building Materials.

University of Pittsburgh

Air Conditioning Requirements for Hospitals.

Princeton University

Studies of Sound Control in Ventilating Systems.

University of Toronto

Insulating Materials Used in Heating and Ventilating Practice.

University of Wisconsin

Aeration of Buildings.
Effect of Entering Air on Temperature, Velocity and Distribution of Air in Enclosed Spaces.
Heat Transfer in Storage Water Heaters.

CONTRIBUTORS TO RESEARCH

Contributions in support of the research work of the Committee on Research are hereby acknowledged from the following:

Allegheny City Steam Heating Co.; American Rolling Mill Co.; Anemostat Corporation of America; The Auer Register Co.; Barber-Colman Co.; Bristol Co.; Buffalo Forge Co.; E. K. Campbell Heating Co.; Carrier Corp.; Detroit Edison Co.; Detroit Lubricator Co.; Excelsior Steel Furnace Co.; Federal Reserve Board Building Management; Julien P. Friez & Sons, Inc.; The Fulton Sylphon Co.; The G & O Manufacturing Co.; General Electric Co.; Hart & Cooley Co.; *Heating, Piping and Air Conditioning Contractors National Association;* Ilg Electric Ventilating Co.; Illinois Testing Laboratories; Independent Register Co.

Johnson Service Co.; Kewanee Boiler Corp.; Lastick Products Co., Inc.; May Oil Burner Corp.; Metropolitan Life Insurance Co.; F. Meyer & Bros.; Milcor Steel Co.; Modine Mfg. Co.; Mueller Brass Co.; *National Association of Woodwork Jobbers, Inc.; National Door Manufacturers Association;* John J. Nesbitt, Inc.; Permutit Co.; Reeves Mfg. Co.; Rochester Gas and Electric Corp.; San Antonio Public Service Co.; *Steel Heating Boiler Institute;* The Trane Co.; Tremco Mfg. Co.; Tuttle & Bailey, Inc.; Union Electric Company of Missouri; U. S. Register Co.; The Waterloo Register Co.; and Westinghouse Electric & Mfg. Co.

The report of the tellers was given by the secretary, Mr. Hutchinson, as follows:

Report of Tellers of Election

BALLOT FOR OFFICERS

	For
President—J. F. McIntire	692
First Vice-President—F. E. Giesecke	691
Second Vice-President—W. L. Fleisher	675
Treasurer—M. F. Blankin	692

Scattered votes were recorded for nine other members.

Members of the Council—Three-Year Term

E. K. Campbell	688
S. H. Downs	690
A. J. Offner	689
G. L. Tuve	690

Scattered votes were recorded for five other members.

Total Ballots Counted	737
Ballots Invalid	36

BALLOT FOR COMMITTEE ON RESEARCH

Three-Year Term

R. E. Daly	691
W. A. Danielson	690
H. J. Rose	689
C. Tasker	691
C.-E. A. Winslow	689

Scattered votes were recorded for five other members.

Total Ballots Counted	737
Ballots Invalid	36

BOARD OF TELLERS

A. C. BUENSOD, *Chairman*
C. S. PABST
E. N. SANBERN

Report of Tellers

BALLOT FOR AMENDMENT TO CONSTITUTION AND BY-LAWS ARTICLE C-V, SECTION 1

Number of Ballots *For*	611
Number of Ballots *Against*	23
Total Number of Ballots	634
Illegal Ballots	29

BOARD OF TELLERS

H. G. MEINKE, *Chairman*
W. E. HEIBEL
E. N. SANBERN

New Officers Installed

Professor Rowley, past president of the Society, conducted the installation, assisted by F. C. McIntosh and Prof. C. M. Humphreys, Pittsburgh, Pa. President-elect J. F. McIntire expressed his appreciation of the honor of being president of the Society. The other officers installed were Prof. F. E. Giesecke, 1st vice-president; W. L. Fleisher, 2nd vice-president, and M. F. Blankin, Philadelphia, Pa., treasurer. The newly elected Council members were introduced, Messrs. E. K. Campbell, S. H. Downs, Kalamazoo, Mich.; A. J. Offner, New York, N. Y., and G. L. Tuve.

President McIntire called for unfinished business and the secretary was requested to read the proposed amendment to the By-Laws relating to initiation fees and dues. The amendments to Article B-IV, Sections 1, 2, 5 and 7 were read. On motion of Mr. Fleisher, seconded by Professor Rowley, it was voted to adopt the amendment to Article B-IV, Section 1.

Article B-IV, Section 1—The admission fee of Members, Associate Members, Junior Members and Student Members shall be as determined by the Council until 1940 and thereafter the admission fee of Members and Associate Members shall be fifteen dollars ($15.00); of Junior Members five dollars ($5.00); and of Student Members two dollars ($2.00). Admission fee must accompany application.

To be amended as follows:

The admission fee of Members, Associate Members, Junior Members and Student Members shall be as determined by the Council until 1942 and thereafter the admission fee of Members and Associate Members shall be fifteen dollars ($15.00); of Junior Members five dollars ($5.00); and of Student Members two dollars ($2.00). Admission fee must accompany application.

On motion of Professor Rowley, seconded by Mr. Fleisher, the following amendment was unanimously adopted, Article B-IV, Section 2.

Article B-IV, Section 2—The annual dues of Members, Associate Members, Junior Members and Student Members shall be as determined by the Council until 1940 and thereafter the dues of Members and Associate Members shall be twenty-five dollars ($25.00); of Junior Members twelve dollars ($12.00); and of Student Members five dollars ($5.00).

To be amended as follows:

The annual dues of Members, Associate Members, Junior Members and Student Members shall be as determined by the Council until 1942 and thereafter the dues of Members and Associate Members shall be twenty-five dollars ($25.00); of Junior Members twelve dollars ($12.00); and of Student Members five dollars ($5.00).

After some discussion by C. S. Leopold, Philadelphia, Pa., Professor Rowley, R. A. Miller, Messrs. Gurney, Fleisher and Offner, on motion of J. J. Aeberly, Chicago, Ill., seconded by Professor Giesecke, it was voted to reject the amendment to Article B-IV, Section 5. On motion of Professor Rowley, seconded by Mr. Aeberly, it was voted to reject the amendment to Article B-IV, Section 7. President McIntire called for discussion of new business and Mr. Gurney offered the following resolution, seconded by L. T. Avery, Cleveland, Ohio:

That a committee shall be appointed for the purpose of studying the entire question of the selection of council members and officers in order that due representation shall

be given to the important number of society members who have no chapter affiliations. I propose that this committee shall consist of the chairman of the Advisory Council, the president, the chairman of the Constitution and By-Laws Committee, the chairman of the Nominating Committee, and two members at large to be nominated now from the floor of this meeting.

Comments were made by Messrs. Offner, Campbell, Wiggs, Downs, W. A. Russell, Waterbury, Conn.; Tom Brown, Chicago; C. H. Randolph, Milwaukee, Wis., and N. D. Adams, Rochester, Minn.

The resolution was unanimously adopted.

President McIntire invited nominations from the floor for the election of two members to serve on a special committee authorized by vote of the meeting. Prof. F. B. Rowley nominated N. D. Adams; Prof. C. M. Humphreys, Pittsburgh, Pa., nominated E. K. Campbell; J. H. Ferguson, Cleveland, Ohio, nominated L. T. Avery; F. C. McIntosh nominated W. T. Jones, Boston, Mass.; Mr. Brown nominated W. A. Russell. These nominations were properly seconded and on motion of Mr. Gurney, seconded by Mr. Wiggs, it was voted that nominations be closed and that a vote by ballot be recorded by the secretary of the Society.

On motion of Mr. Gurney, seconded by Mr. Miller, it was voted that the president of the Society or his appointee act as chairman of the special committee.

Mr. Gurney then offered a petition for a revision of the By-Laws signed by 10 members, which under Article B-XVI of the By-Laws would be submitted to all members at least 30 days in advance of the next Society meeting.

Resolutions

H. M. Hendrickson, Los Angeles, Calif., offered the following resolutions for adoption:

Whereas, the 45th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS has been a great success, it is fitting that we should express our appreciation and thanks:

To the Pittsburgh Chapter, for the excellent manner in which the arrangements for this meeting were handled;

To the members of the committees of the Pittsburgh Chapter, for the excellent management of the meeting and their friendly hospitality.

To the City of Pittsburgh, its citizens and its officials, for their hospitality;

To the William Penn Hotel, its employees, and management, for the excellent manner in which they have handled arrangements and the service which they have given us during our stay here;

To the Pittsburgh Convention and Tourist Bureau for their assistance and interest;

To the speakers and the authors of the many technical papers and for their excellent presentation of the material that has been so valuable to us here;

To the newspapers and the trade journals for the excellent publicity which they have given this meeting;

To the personnel of the Headquarters office staff, for the manner in which they have helped us and made us feel at home;

To the national officers of the Society, for the smooth manner in which all the meetings and events were handled.

And last, but not least, we thank the ladies for coming here and for their gracious presence, which we appreciate.

The motion was seconded by W. H. Badgett, College Station, Tex.

Mr. Miller, president of the Pittsburgh Chapter, responded to the resolution and thanked the members present for their attendance at the meeting, for the cooperation given by everyone, and said that the success of the 45th Annual Meeting of the Society had been due to the complete and enthusiastic acceptance of both the technical and entertainment programs by the visiting members.

The meeting then adjourned.

Entertainment

Visiting members, guests, and ladies, who attended the 45th Annual Meeting of the Society, took up the program of entertainment which was planned by the Pittsburgh Chapter with a spirit and zest which assured the success of the various scheduled events.

At 3:00 p.m. on Sunday, January 22, Pres. E. Holt Gurney was guest speaker on the Armco Hour, broadcast over radio station KDKA at Pittsburgh and the NBC Blue Network. Arrangements were made to receive the broadcast in the Pittsburgh Chapter Room of the William Penn Hotel, where the reception was held.

Chapter Delegates' Conference

The Second Conference of Chapter delegates was held at the William Penn Hotel on Monday morning, January 23. Pres. E. Holt Gurney presided and among those attending were: T. T. Tucker, Atlanta; E. B. Royer, Cincinnati, G. M. Simonson, Golden Gate; J. R. Vernon, Illinois; W. R. White, Iowa-Nebraska; C. A. Flarsheim, Kansas City; William Worton, Manitoba; James Holt, Massachusetts; F. J. Linsenmeyer, Michigan; W. W. Bradfield, Western Michigan; J. E. Swenson, Minnesota; F. J. Friedman, Montreal; H. G. Meinke, New York; J. J. Landers, Western New York; C. M. H. Kaercher, Northern Ohio; E. W. Gray, Oklahoma; H. R. Roth, Ontario; R. O. Wesley, Pacific Northwest; H. H. Erickson, Philadelphia; C. M. Humphreys, Pittsburgh; E. E. Carlson, St. Louis; H. M. Hendrickson, Southern California; W. H. Badgett, South Texas; S. P. Eagleton, Washington, D. C.; H. C. Frentzel, Wisconsin; T. H. Anspacher, North Texas. Others present were Tom Brown, Chicago, G. L. Wiggs, Montreal, E. F. Dawson, Oklahoma City, R. C. Bolsinger, Philadelphia, W. A. Russell, Waterbury, Conn., W. G. Boales, Detroit, and L. H. Laffoley, Montreal.

At 5:00 p.m. on Tuesday, January 24, a special broadcast from station KDKA was arranged by T. F. Rockwell, secretary of Pittsburgh Chapter, on the Relationship of Atmospheric Environment to Comfort and Health. Those who took part in this radio interview were President Gurney, W. L. Fleisher, Prof. C.-E. A. Winslow, Dr. Charles Sheard and Mr. Sutherland, KDKA announcer.

The past presidents dined at 6:30 p.m. and those who enjoyed this annual reunion were Homer Addams, F. B. Rowley, John Howatt, W. T. Jones, J. F. Hale, G. L. Larson, Thornton Lewis, W. H. Carrier, H. P. Gant, C. V. Haynes and W. H. Driscoll.

This year one of the outstanding events of the meeting took place at the Society's annual banquet on Wednesday night in the grand ballroom, when Thornton Lewis presented the F. Paul Anderson Medal to Prof. F. B. Rowley. This award was created by Mr. Lewis in tribute to the late Dean Anderson and has been given on only two previous occasions—in 1932 to W. H. Carrier and in 1936 to Dr. A. C. Willard. It was awarded to Professor Rowley for his attainments in the field of thermal research and their influence upon the arts of heating, ventilating and air conditioning and for his outstanding services to the Society as a member and officer. Upon receiving the medal, Professor Rowley expressed his appreciation for this award. The past president's emblem was presented to President Gurney by W. T. Jones. Prof. E. O. Eastwood, who was toastmaster, introduced Dr. John Gabbert Bowman, chancellor of the University of Pittsburgh. Dr. Bowman gave an inspiring address on the desirable balance to be obtained in life between an appreciation of culture and beauty and scientific knowledge. Mr. Gurney spoke briefly of his pleasure at having served the Society as president and introduced J. F. McIntire, president elect, who responded. Dancing followed the banquet in the Urban Room and the music was by Joey Sims and his orchestra.

Members and guests left Pittsburgh with a feeling that they had participated in a very enjoyable and successful meeting and offered their thanks and congratulations to the Committee on Arrangements.

Committee on Arrangements

R. A. MILLER, President Pittsburgh Chapter; J. F. S. COLLINS, JR., *General Chairman;* ARTHUR MCGONAGLE, *Honorary Chairman;* C. M. HUMPHREYS, *First Vice-Chairman;* F. C. MCINTOSH, *Second Vice-Chairman.*

Finance Committee: M. L. Carr, *Chairman;* L. S. Maehling; E. H. Riesmeyer, Jr.; R. J. J. Tennant; R. A. Miller; E. C. Smyers.

Publicity Committee: T. F. Rockwell, *Chairman;* F. B. Mahon; R. B. Stanger.

Attendance Committee: A. F. Nass, *Chairman.*

Reception Committee: J. E. Frazier, *Chairman;* Mrs. H. A. Canon, *Vice-Chairman;* H. A. Beighel; H. A. Biber; G. C. Blackmore; Roy Brauer; E. M. Burkhart; G. M. Comstock; R. P. Dickinson, Jr.; R. W. Dickson, Jr.; T. M. Dugan; J. E. Eckstein; L. C. Eils; G. P. Ellis; F. H. Gallagher; C. B. Hathaway; F. H. Hecht; R. H. Heilman; F. C. Houghten; E. H. Hyde; P. M. Kimmel; B. D. Landes; Walter Lowe; F. E. Lynn; J. E. Mueller; P. Nicholls; H. E. Park; G. W. Penney; John Proie; I. G. Reed; Van A. Reed, Jr.; E. S. Scanlon;

E. T. Selig, Jr.; L. T. Sherwood; Charles Sonneborn; W. W. Stevenson; F. A. Sutherland; G. O. Weddell; Mrs. H. A. Biber; Mrs. G. P. Ellis; Mrs. J. E. Frazier; Mrs. H. E. Lore; Mrs. D. W. Loucks; Mrs. F. B. Mahon; Mrs. J. E. McLean; Mrs. H. Lee Moore; Mrs. W. H. Reed, III; Mrs. E. C. Smyers; Mrs. P. C. Strauch; Mrs. R. J. J. Tennant; Mrs. G. G. Waters.

Technical Sessions Committee: P. A. Edwards, *Chariman;* Leighton Orr, *Vice-Chairman.*

Transportation Committee: D. W. Loucks, *Chairman;* H. A. Canon; G. G. Waters.

Inspection Committee: R. J. J. Tennant, *Chairman;* F. N. Speller.

Ladies Committee: Mrs. J. F. S. Collins, Jr., *Chairman;* Mrs. F. C. Houghten, *Vice-Chairman;* Mrs. C. M. Humphreys; Mrs. R. A. Miller; Mrs. F. C. McIntosh; Mrs. A. F. Nass; Mrs. Leighton Orr; Mrs. R. B. Stanger; Mrs. T. F. Rockwell; Mrs. R. J. J. Tennant.

Entertainment Committee: R. B. Stanger, *Chairman;* H. A. Canon; C. M. Humphreys; H. Lee Moore; R. J. J. Tennant.

Banquet Committee: E. C. Smyers, *Chairman;* J. E. McLean, *Vice-Chairman;* Mrs. R. A. Miller; W. A. Allen; K. C. Lewis; H. E. Lore; A. W. Marshall; C. E. Parks; P. C. Strauch; W. H. Reed, III.

PROGRAM 45TH ANNUAL MEETING

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
WILLIAM PENN HOTEL, PITTSBURGH, PA.
JANUARY 22-26, 1939

COMMITTEE MEETINGS

Sunday, January 22

10:00 A.M. Committee on Research—Parlors B and C
2:00 P.M. Council Meeting (Forum Room)
2:30 P.M. Committee on Air Conditioning Requirements of Glass—Parlor D

Monday, January 23

9:00 A.M. Committee on Air Distribution and Air Friction (Parlor B)
10:00 A.M. Committee on Corrosion in Steam Systems (Parlor C)
5:00 P.M. Committee on Radiation and Comfort (Parlor D)

Tuesday, January 24

12:00 NOON Luncheon, Committee on Sensations of Comfort (Parlor B)

Wednesday, January 25

9:00 A.M. Committee on Air Cleaning and Atmospheric Impurities (Parlor D)
12:00 NOON Nominating Committee Luncheon (Forum Room)
12:00 NOON Luncheon, Committee on Solid Fuels (Parlor C)

Thursday, January 26

12:30 P.M. Council Meeting (Forum Room)

TECHNICAL SESSIONS

Monday, January 23

8:30 A.M. Registration—Pittsburgh Chapter Room (17th floor)

9:30 A.M. Meeting of Chapter Delegates (Forum Room, 17th floor)

2:00 P.M. Afternoon Session (Grand Ball Room, 17th floor). President E. Holt Gurney, Presiding

Call to order
Announcements
Reports of Officers
Reports of Council Committees
Report of Committee on Research, W. L. Fleisher
Introduction of Past Chairmen of Committee on Research

Technical Papers:

Frictional Resistance to the Flow of Air in Straight Ducts by F. C. Houghten, J. B. Schmieler, John Zalovcik and Nicholas Ivanovic
Air Conditioning in Industry by W. L. Fleisher, A. E. Stacey, Jr., F. C. Houghten and M. B. Ferderber
Report of Tellers of Election

Tuesday, January 24

9:30 A.M. Morning Session (Grand Ball Room, 17th floor). President E. Holt Gurney, Presiding

Physiological Sensations:

Recent Advances in Physiological Knowledge and Their Bearing on Ventilation Practice by C.-E. A. Winslow.
Cardiac Output, Peripheral Blood Flow, and Blood Volume Changes in Normal Individuals Subjected to Varying Environmental Temperatures by F. K. Hick, R. W. Keeton, Nathaniel Glickman and H. C. Wall
The Role of the Extremities in the Dissipation of Heat from the Body in Various Atmospheric and Physiological Conditions by Charles Sheard, Marvin M. D. Williams, Grace M. Roth and Bayard T. Horton
Skin Temperatures of the Extremities and Effective Temperature, by Charles Sheard, Marvin M. D. Williams and Bayard T. Horton
Air Conditioning Requirements of an Operating Room and Recovery Ward by F. C. Houghten and W. Leigh Cook, Jr.

2:00 P.M. Afternoon Session (Grand Ball Room, 17th floor). 1st Vice-Pres., J. F. McIntire, Presiding

Heat Transfer:

Some Reflection and Radiation Characteristics of Aluminum, by C. S. Taylor and J. D. Edwards
A Study of the Heat Requirements of a Single-Glazed Test House and a Double-Glazed Test House by M. L. Carr, R. A. Miller, Leighton Orr and David Shore
Heat Transfer in Storage Water Heaters by D. W. Nelson and A. A. Rosenberg
Condensation of Moisture and Its Relation to Building Construction and Operation by F. B. Rowley, A. B. Algren and C. E. Lund

Wednesday, January 25

9:30 A.M. Morning Session (Urban Room, 17th floor), 1st Vice-Pres., J. F. McIntire, Presiding

Fuels:

Responsibility of ASHVE Towards Solid Fuel Industry by W. A. Danielson
Selection of Solid Fuels from the Viewpoint of the Small Consumer by P. Nicholls.
Small Stokers by P. A. Mulcey and R. A. Sherman.
Performance of Stoker-Fired and Hand-Fired Warm-Air Furnaces in the Research Residence by A. P. Kratz, S. Konzo and R. B. Engdahl

2:00 P.M. Afternoon Session (Urban Room, 17th floor). 2nd Vice-Pres., F. E. Giesecke, Presiding

Air Pollution:

Smoke Abatement—Where to Draw the Line by H. B. Meller
Smoke Producing Tendencies in Coals of Various Ranks by H. J. Rose and F. P. Lasseter
Air Filter Performance as Affected by Low Rate of Dust Feed, Various Types of Carbon, and Dust Particle Size and Density by F. B. Rowley and R. C. Jordan

Thursday, January 26

9:30 A.M. Morning Session (Urban Room, 17th floor). President E. Holt Gurney, Presiding

Installation of New Officers

Design Factors:

The Effect of Heat Storage and the Variation in Outdoor Temperature and Solar Intensity on the Heat Transfer Through Walls by J. S. Alford, J. E. Ryan and F. O. Urban
Analysis of Weather Design Data in the United States by J. C. Albright
Application of Summer Weather Data in Design by John Everetts, Jr.
The Flow of Heat Through Walls by F. E. Giesecke
Reactions of Office Workers to Air Conditioning in South Texas by A. J. Rummel, F. E. Giesecke, W. H. Badgett and A. T. Moses
Unfinished Business
New Business
Resolutions
Adjournment

ENTERTAINMENT

Sunday, January 22

2:00 P.M. Informal Reception of Visiting Members and Ladies, Pittsburgh Chapter Room (17th floor, William Penn Hotel)

3:00 P.M. Radio Broadcast by President E. Holt Gurney, Guest Speaker on the Armco Hour, Station KDKA Pittsburgh and NBC Blue Network Stations (arrangements have been made to receive this broadcast in the Pittsburgh Chapter Room)

7:00 P.M. to 10:00 P.M. Informal Reception Continued

Monday, January 23

12:15 P.M. Welcome Luncheon—Members, Guests and Ladies
Greetings by Mayor Cornelius D. Scully of Pittsburgh

Address by Charles E. Mayette, The Position of the Consulting Engineer in the U. S. Housing Problem

3:00 P.M. Ladies Tea (Forum Room, 17th floor)

5:00 P.M. Social Hour (Pittsburgh Chapter Room, 17th floor)

7:00 P.M. Jamboree Dinner, The Chatterbox, Lower Lobby Floor—William Penn Hotel. Dinner will be served at popular prices—Order what you like—Regular Dinners or a la Carte.

Pittsburgh Chapter presents special entertainment program—Dancing without additional charge. Hour and a half Floor Show—Music by Little Jackie Heller and his orchestra

Tuesday, January 24

1:00 P.M. Ladies' Bridge Luncheon (Cardinal Room, 17th floor)

5:00 P.M. Special Broadcast from Station KDKA—Relationship of Atmospheric Environment to Comfort and Health

5:15 P.M. The film, "Steel—Man's Servant" in technicolor will be shown, Edwin C. Hill is the narrator and the musical score is directed by Robert Armbruster. The thrilling drama of steel making is dramatically told in 40 minutes.

6:30 P.M. Past Presidents' Reunion Dinner (Parlor B)

9:30 P.M. A Night in Bavaria—Entertainment by World Renowned Bavarians, William Penn Bavarian Room. Special Supper for ASHVE Members.

Wednesday, January 25

10:00 A.M. Ladies' Tour to East Liberty Presbyterian Church, the Schenley District Civic Center and Luncheon

7:00 P.M. Annual Banquet and Dance (Grand Ball Room, 17th floor). Presentation of F. Paul Anderson Medal to Prof. Frank B. Rowley. Presentation of Past Presidents Emblem to E. Holt Gurney. Prof. E. O. Eastwood, Toastmaster. Address by Dr. John Gabbert Bowman, Chancellor of University of Pittsburgh

Thursday, January 26

10:00 A.M. Ladies:—Reserved for Shopping

1:00 P.M. Inspection Trips—Sign-up at Registration Desk

No. 1105

FRICTIONAL RESISTANCE TO THE FLOW OF AIR IN STRAIGHT DUCTS

By F. C. HOUGHTEN,* J. B. SCHMIELER,** (*MEMBERS*), J. A. ZALOVCIK,*** AND N. IVANOVIC,† PITTSBURGH, PA.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in its Research Laboratory at the Pittsburgh Experiment Station of the U. S. Bureau of Mines.

INTRODUCTION BY J. H. VAN ALSBURG

This paper presents the results of work conducted at the Society's Laboratory under a program supervised by the Technical Advisory Committee on Air Distribution and Air Friction. The purpose of the investigation was to obtain accurate and more complete data on the flow of air in duct systems. A comprehensive search of all published literature on the subject was made to determine what reliable research work had been done on air or other fluids, since fundamentally the basic relations apply alike to all fluids whether liquid or gaseous under the relatively low pressure conditions in which we are particularly interested.

The first part of this work necessarily had to be confined to the fundamental relationship for the simplest elements of a duct system, and additional phases of the work will be undertaken in the order of their importance so that the more complicated arrangements found in practical work will be included eventually.

While this first paper gives the results of the Society's Laboratory work to date, it must be realized that these data are not complete enough to be directly applicable in estimating resistances of duct systems so that other papers to follow will give supplementary or revised information when further laboratory work has been completed. Ultimately it is expected that sufficient data will be obtained from this program to permit the recommendation of a method of computing the resistances of complete duct systems for inclusion in the Heating, Ventilating and Air Conditioning Guide.

THE increase in the use of air distributing systems in connection with heating, cooling and air conditioning during the past decade has resulted in a more critical interest in the design and cost of duct work. In this connection, there developed a lack of confidence in the available design data for estimating the carrying capacity of a given size of duct for a given pressure loss. In order to correct this condition, the Research Laboratory was asked to make a study of the frictional resistance to the flow of air in ducts and fittings.

* Director, ASHVE Research Laboratory.
** Research Engr., ASHVE Research Laboratory.
*** Research Asst., ASHVE Research Laboratory.
† Research Asst., ASHVE Research Laboratory.

Presented at the 45th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Pittsburgh, Pa., January, 1939.

The work was undertaken at the Laboratory under the Technical Advisory Committee on Air Distribution and Air Friction, whose personnel during 1938 included: J. H. Van Alsburg, *Chairman*, S. H. Downs, M. K. Fahnestock, F. J. Kurth, R. D. Madison, L. G. Miller, D. W. Nelson, C. H. Randolph, Ernest Szekely and G. L. Tuve.

The program originally laid out by the Committee included a study of the frictional resistance to the flow of air in straight runs, elbows, Y's, reducers, and other fittings commonly included in air distributing systems. The study

FIG. 1. TEST SET-UP SHOWING 7.2-IN. SQUARE FACTORY-MADE DUCT. OTHER DUCT SET-UPS WERE ESSENTIALLY THE SAME

of elbows was to include variations in radius of curvature, with and without turning blades. The first set-up was designed to include two elbows with variations in the set-up to include Y's and other connections. Early in the investigation difficulty was experienced in accounting for some of the results obtained both for the straight ducts and more particularly for the elbows, and in order to simplify the procedure attention was limited to the study of the flow of air in straight ducts. The study of straight ducts was outlined to include round ducts, square ducts, and ducts of rectangular cross-section of various aspect ratios. Different essentials in design and workmanship including ducts laid out and built in local shops and ducts fabricated under mass production methods in a factory for wide distribution were to be studied. Here

again, the difficulties involving variations in joints due to design and workmanship resulted in a further curtailment of the original program to include only round and square ducts of two designs, referred to as *shop-made* and *factory-made* with the exception of a single rectangular duct, 15$\frac{5}{16}$ by 3$\frac{1}{4}$ in. in cross-section. The future program of the Committee provides for a continued investigation of those phases of the study not completed to date including rectangular ducts, and fittings.

This paper includes the results of the study on the frictional resistance to the flow of air in 4-in., 8-in. and 24-in. round ducts and square ducts of approximately equivalent carrying capacity for the same frictional resistance. Results of the frictional resistance to the flow of air in straight ducts without joints and the added resistance offered by joints are included. The paper also presents the relationship between the friction factor and the Reynolds Number found for the round ducts. A modified form of the Fanning equation is shown to apply with a high degree of accuracy for the three sizes of ducts, justifying its use for expanding friction flow characteristics from the three sizes of ducts studied to smaller, intermediate and larger sizes, which can be used with confidence for ducts up to at least 4 ft in diameter, and probably without any considerable error for a wider range. Friction charts based on this expansion are presented for round duct without joints and for round duct with 40 joints per 100 ft of run, ranging from 3 in. to 100 in. in diameter. Based upon the relation between the friction found for round and for square duct and assuming spacing of joints as found in common practice, a modified form of the conversion equation was developed for determining the diameter of a round duct having the same carrying capacity with the same frictional resistance as a given rectangular duct, and a conversion table based upon this formula is presented.

TEST SET-UP AND PROCEDURE

Fig. 1 is a photograph of the 7.2-in. square factory-made duct and Fig. 2 a drawing of the test set-up as used for the 8-in. round, shop-made duct. Excepting for transformation pieces to accommodate the different sizes of duct and a larger fan used in the case of the 24-in. round and 22-in. square ducts, essentially the same arrangement was used in studying the other sizes. Elbow losses were studied in connection with the 4- and 8-in. ducts. Since this phase of the study was postponed, elbows were not included in the set-up for the 24-in. round and equivalent square ducts. The desired air velocity through the test section was supplied by a blower equipped with a throttling damper on the intake side. Just beyond the outlet of the blower a relief damper, consisting of twenty-six 1-in. holes around the duct and equipped with a collar, gave further control of the rate of air delivery. For ease in measuring the desired rate of air flow, a U. S. Bureau of Standards calibrated nozzle was located between two 10-ft sections of 12-in. duct. A test section of 8-in. duct was connected to the 12-in. measuring section through a 2-ft, 5-deg transformation piece, a 4-ft section of 8-in. duct containing a straightener, and a 10-ft section containing a Pitot tube. A straightener also preceded the measuring section of 12-in. duct.

For all sizes of duct studied approximately 60 ft of straight run was used. Provision was made for observing the wet- and dry-bulb temperature of the

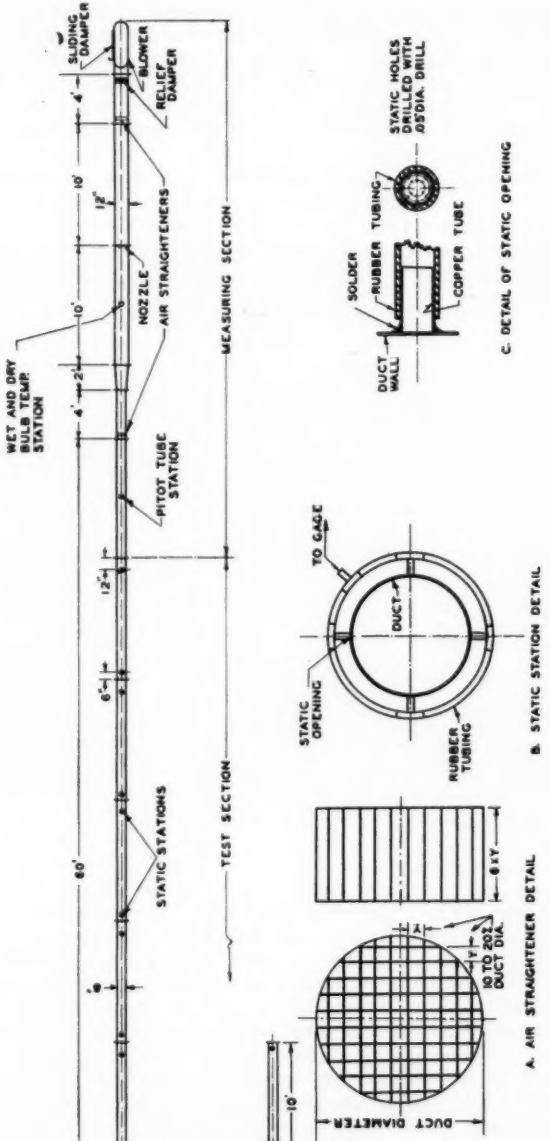

FIG. 2. TEST SET-UP SHOWING 8-IN. ROUND DUCT. OTHER DUCT SET-UPS WERE ESSENTIALLY THE SAME

air after it passed through the nozzle. A Pitot tube station was located in the 60-ft straight run of 8-in. duct 7½ diameters beyond the straightener. A number of static stations for measuring pressure were located throughout the length of the test section. While the nozzle referred to previously was always used to observe the rate of air flow, the velocity was also checked by a standard Pitot tube whose design is specified in the 1938 edition of the Standard Test Code for Centrifugal and Axial Fans, adopted by a Joint Fan Test Code Committee of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *National Association of Fan Manufacturers.* Three other Pitot tubes were used. It is of interest to note that, with the precautions taken to

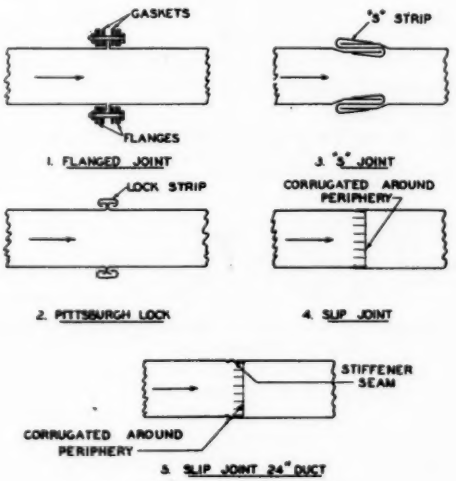

FIG. 3. DETAIL OF THE DIFFERENT JOINTS USED

insure uniform flow, of the four Pitot tubes used one read about 0.6 per cent lower, another about 0.8 per cent lower, and a Prandtl tube gave identical results as the standard tube. The standard Pitot tube gave air quantities 1.1 per cent larger than the calibrated nozzle.

Static stations were placed at frequent intervals along the test section of the duct. The type of static pressure opening is shown in detail in Fig. 2C, and the arrangement whereby four static pressure openings were connected into one tube is shown in Fig. 2B. The type of air straightener used was of the *egg-crate* design and is shown in Fig. 2A. This arrangement was adopted after some experimentation and a great deal of consultation with other experimenters interested in this type of work.

The static pressure differences between stations along the duct, for the Pitot tubes, and for the nozzle, were obtained by the use of either an inclined

manometer gage or a Wahlen gage, the latter being used when the pressure differences were small. Xylene with aniline dye added was used in the inclined gage because its density does not vary appreciably with temperature and evaporation. The Wahlen gage[1] used is accurate to 1/1000 of an inch of water pressure.

The formula used to give the air delivery by the nozzle was:

$$V = 4005\, C_d C_n \sqrt{h/S} \quad \ldots\ldots (1)$$

where,

V = actual nozzle velocity in feet per minute
h = pressure drop across the nozzle in inches of water
C_d = coefficient of discharge = 0.98

$$C_n = \text{nozzle coefficient} = \frac{1}{\sqrt{1 - (d_2/d_1)^4}}$$

where,

d_2 = throat diameter of nozzle
d_1 = line diameter of nozzle
S = ratio of test air density to standard air density (dry air at 70 F and 29.92 in. Hg) $= \dfrac{17.72 \times BP - 6.70 \times E}{460 + T_t}$

where,

BP = barometric pressure in inches of Hg
E = vapor pressure at the dew-point of the mixture
T_t = test temperature in degrees Fahrenheit.

To obtain the velocity in the duct the following relation was used:

$$V_d = \frac{V_n A_n}{A_d} \quad \ldots\ldots (2)$$

where,

V_d = duct velocity
V_n = nozzle velocity
A_n = nozzle throat area
A_d = duct area.

The Pitot tube readings were converted into velocities by using the formula:

$$V = 4005\sqrt{h/S} \quad \ldots\ldots (3)$$

where,

h = velocity head in inches of water
S = same as in formula (1) for air delivery by the nozzle.

In making a traverse the positions at which observations were made were such as to give an average velocity when the arithmetical average of the several velocities was taken. The positions were determined by dividing the duct area into several equal concentric areas. The position taken in each concentric area was such that a circle drawn through it would divide this area into two equal concentric areas. For any circle thus drawn there would be

[1] See Bibliography (1).

two observation points along any diameter, one on either side of the center of the duct. It can be seen that the arithmetical average of the velocities at these points approximates the average velocity in the duct. This approximation improves as the number of points is increased. A mathematical equation can be used giving the distance along the radius where readings are to be taken for a duct divided into any number of equal concentric areas. This equation is:

$$r = \frac{D}{2}\sqrt{\frac{2X-1}{2N}} \qquad (4)$$

where,

r = distance along a radius from the center of the duct to the point at which the reading is to be taken

D = diameter of duct

X = number of the concentric area where the central area is 1

N = number of equal concentric areas into which the duct area is to be divided.

Average duct velocities were also obtained by plotting the velocities at these points, integrating the area with a planimeter and calculating average velocities. This method was found to give slightly better results and was used for actual calculations.

Before making a test on any given duct the intake and relief dampers were arranged to give the desired rate of flow, after which the system was continued in operation long enough to establish equilibrium of temperature and flow before making observations. The temperature (dry-bulb and wet-bulb) of the air, rate of air flow, barometric pressure and pressure drops along the duct were then taken by one observer while an assistant watched the nozzle differential gage in order to insure uniform rate of air flow during the observations. In most tests, several pressure drops between as many different pairs of static stations were observed in order to give pressure drop flow relationship through different parts of the duct, particularly as it was affected by variations in joints. The pairs of static pressure stations were chosen so as to include the pressure drop through the maximum length of straight duct including all joints, the maximum length of straight duct not including any joint, and the drop across a single joint. A series of tests was always made on a given test set-up at velocities ranging from the minimum giving a satisfactory pressure difference for the given size of duct, up to the maximum possible with the fan and duct arrangement, which was usually about 3500 fpm. Check tests were always made on a given set-up on at least two different days. If these results did not check, the system was carefully examined for defects, usually leaks, and tests were repeated until satisfactory results were obtained. Joint leaks were prevented by first sealing with plasticine and then wrapping with adhesive tape which was painted aluminum.

TEST RESULTS

Results of a great many tests on different set-ups, including shop-made and factory-made round duct 8 in. in diameter, shop and factory-made square duct of approximately the same carrying capacity, and a factory-made rectangular

duct of approximately the same carrying capacity are given in Fig. 4. Curves with test points are given for 7¼-in. square shop-made duct with average joints, 7.2-in. square factory-made duct with average joints, 8-in. round shop and factory-made duct with average joints, and 15 5/16 by 3¾-in. factory-made with average joints. The 8-in. round, and equivalent square and rectangular ducts were the first studied, and a great many tests were made while the effect of joints was being investigated. As a result, these ducts were erected and dismantled a number of times, which accounts for some of the dispersion of test points from the curves as drawn. Nevertheless, the points indicate a very definite curve in each instance. It will be noted that the several ducts as

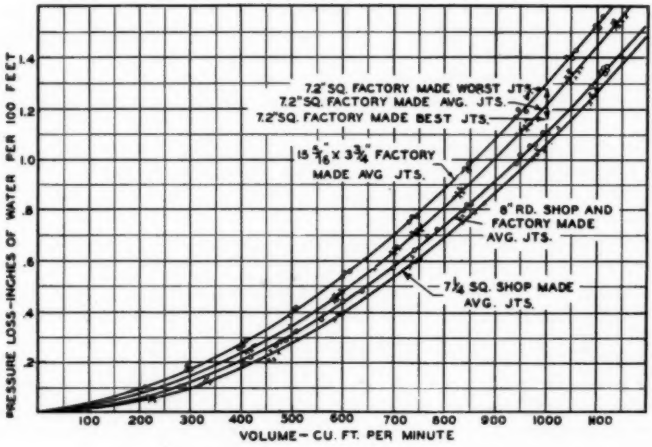

FIG. 4. RELATION BETWEEN PRESSURE LOSS AND VOLUME FOR 8-IN. ROUND AND EQUIVALENT SQUARE AND RECTANGULAR DUCTS WITH ACTUAL TEST POINTS. THE VARIATION IN THE EFFECT CAUSED BY THE BEST AND WORST JOINTS FOUND FOR THE FACTORY-MADE 7.2-IN. SQUARE DUCT IS INDICATED BY ARROWS. THE 7¼-IN. SQUARE SHOP-MADE DUCT HAD 12 JOINTS PER 100-FT LENGTH OF RUN. ALL OTHERS 10 JOINTS

originally chosen did not have the same carrying capacity. The higher resistance shown by the factory-made, square and rectangular ducts was probably due to the relatively crude joints between sections, offering in some cases noticeable constrictions.

Reference to Table 1, which gives the characteristics of the different ducts studied, shows that the same joint distribution was not obtained for all the different ducts. Also, there will be apparent to individuals accustomed to duct design the fact that the length of duct between joints as tested does not in all cases represent common practice. This discrepancy, however, is corrected in later charts, where the resulting calculated relationship, between friction and flow, places all of the data on a more comparable basis. As pointed out previously, there was considerable difference in the uniformity in appearance of

TABLE 1—CHARACTERISTICS OF DUCTS STUDIED

SHAPE OF DUCT	SIZE	MAKE	TYPE JOINT (A)	ACTUAL NO. OF JTS. PER 100 FT.
ROUND	4"	SHOP MADE	4	10
ROUND	4"	FACTORY	4	10
SQUARE	3¾"	SHOP MADE	2&3 (B)	12
ROUND	8"	SHOP MADE	1	10
ROUND	8"	FACTORY	4	10
SQUARE	7¼"	SHOP MADE	2&3 (B)	12
SQUARE	7.2"	FACTORY	3	10
RECTANGULAR	5¼" x 3¾"	FACTORY	3	10
ROUND	24"	SHOP MADE	5	40
SQUARE	22"	SHOP MADE	2	25

A. See Fig. 3.
B. Each joint consisted of No. 2 on two opposite sides and No. 3 on the other two sides.

constrictions resulting from the joints in the different ducts. The 7.2-in. square factory-made and equivalent rectangular ducts showed the most variation in individual joints. The result of this variation on the resistance to flow through the joints of the 7.2-in. square factory-made duct is indicated by arrows on the curve for the duct in Fig. 4.

Curves similar to those in Fig. 4 were plotted for each of the three sizes of duct studied. However, since the length of section between joints did not represent usual practice in all cases, these are not presented, but, instead, there are included Figs. 5, 6 and 7, which give the calculated friction flow relationship for the number of joints per 100-ft length of run indicated in each case

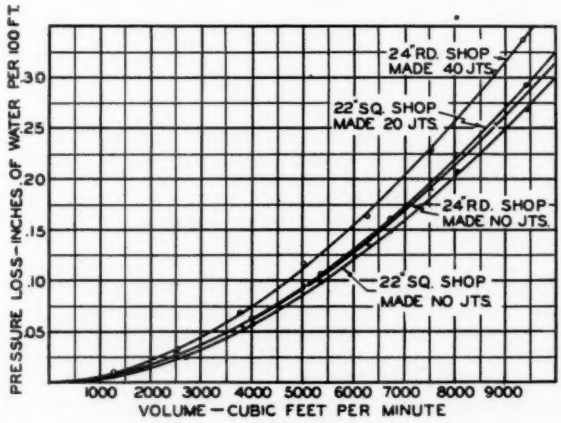

FIG. 5. RELATION BETWEEN PRESSURE LOSS AND VOLUME FOR 24-IN. ROUND AND EQUIVALENT SQUARE DUCT CALCULATED FROM TEST DATA FOR 100-FT LENGTH OF RUN WITH NO JOINTS AND WITH THE INDICATED NUMBER OF JOINTS

for the three sizes of duct tested. For this calculation the frictional resistance to the flow through a duct with no joints was determined by the pressure loss through a test section with no joint between static stations. The frictional resistance through the average joint is given by the formula:

$$J = \frac{h_w - C_o \times L}{N} \quad \ldots\ldots (5)$$

where:

J = frictional resistance through the average joint in the test section
h_w = frictional resistance through the entire test section including straight run of duct and joints
C_o = frictional resistance per foot of duct without joints as defined previously
N = number of joints between static stations
L = total length of test section.

The frictional resistance for 100-ft length of the same duct with any other number of joints, N', then becomes $H_F = 100\ C_0 + JN'$. The number of joints indicated for each size and type of duct is assumed to represent usual practice in each case. The curves mentioned show clearly the variations existing, when results based on common practice using present conversion formulae are plotted. Better results would be obtained if equivalent square and rectangular duct sizes had been calculated using the conversion formula as given later in the paper.

In order to present a better picture of the relation between friction and velocity of flow for the three sizes of round duct, these relationships are all plotted to the same scale in Fig. 8. The relation between the friction through the average joint for each duct and the velocity of flow is given in Fig. 9. The length of straight duct offering frictional resistance equal to that of a single joint in the duct should undoubtedly bear some definite relation to the velocity through the duct. Stated mathematically this equivalent length is defined as:

$$E = \frac{J}{H_f} \quad \ldots\ldots (6)$$

where,

E = equivalent length of joint in feet of duct
J = average joint loss
H_f = static pressure loss per foot of duct without joints

Sufficient data are not available to establish this relationship for all velocities. It is, however, given for a velocity of 2000 fpm in each duct. It will be noted that the 7.2-in. square factory-made joint gives a high friction compared to the other ducts of this equivalent size. The average joints in the different sizes offered resistance to flow at 2000 fpm equal to from a little under ½ ft of straight duct to a little over 2 ft. The high friction offered by the average joint in the 7.2-in. square factory-made duct accounts in part, at least, for the high resistance for this duct with average joints. This was in part due to the many tests made on this duct requiring the duct to be taken apart and reassembled a number of times.

REYNOLDS NUMBER RELATIONSHIP

The relation between the Reynolds Number and the pressure producing flow of any fluid, either liquid or gas, has been a great aid in developing a comprehensive understanding of the flow of such fluids. These relationships have their bearing not only on an understanding of the flow of fluids in pipe, but also on the flow of fluids under any other conditions, including convection currents set up near any heating or cooling surface and the effect of these

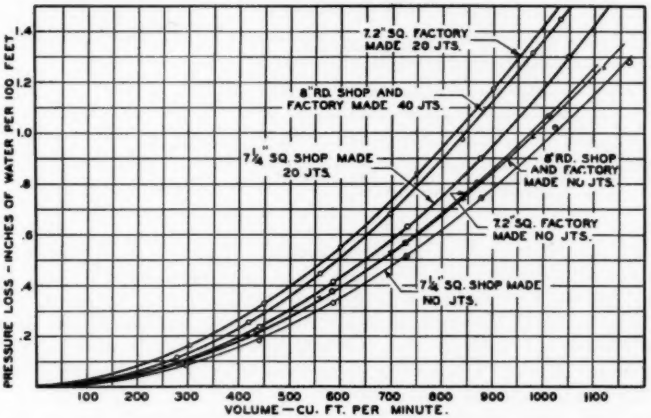

FIG. 6. RELATION BETWEEN PRESSURE LOSS AND VOLUME FOR 8-IN. ROUND AND EQUIVALENT SQUARE DUCT CALCULATED FROM TEST DATA FOR 100-FT LENGTH OF RUN WITH NO JOINTS AND WITH THE INDICATED NUMBER OF JOINTS

convection currents on heat transfer. When applied to the flow of fluids in pipe, the Reynolds Number, defined as a dimensionless number, is expressed as:

$$R = \frac{DV\rho}{\mu} \qquad (7)$$

where:

R = the Reynolds Number
D = diameter of the pipe in feet
V = velocity in feet per second
ρ = density of the fluid in pounds per cubic foot
μ = absolute viscosity of the fluid in pounds per foot second.

Reynolds was able to show that for any pipe and any fluid, either liquid or gas, the flow was streamline when this dimensionless number had a small numerical value, usually below 2000 or 3000, while the flow generally became turbulent when the Reynolds Number exceeded this value. Streamline flow may exist for a Reynolds Number as high as 50,000 if care is taken to

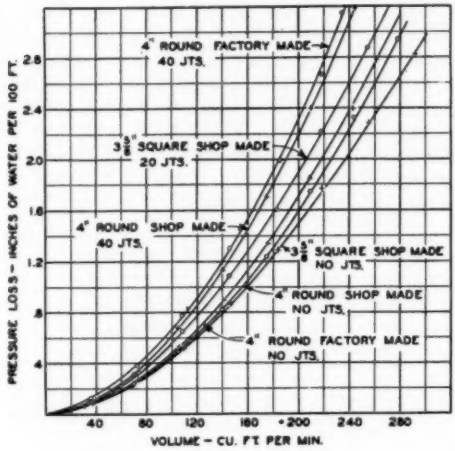
4" ROUND FACTORY MADE 40 JTS.
3 3/8" SQUARE SHOP MADE 20 JTS.
4" ROUND SHOP MADE 40 JTS.
3 3/8" SQUARE SHOP MADE NO JTS.
4" ROUND SHOP MADE NO JTS.
4" ROUND FACTORY MADE NO JTS.
PRESSURE LOSS - INCHES OF WATER PER 100 FT.
VOLUME - CU. FT. PER MIN.

FIG. 7. RELATION BETWEEN PRESSURE LOSS AND VOLUME FOR 4-IN. ROUND AND EQUIVALENT SQUARE DUCT CALCULATED FROM TEST DATA FOR 100-FT LENGTH OF RUN WITH NO JOINTS AND WITH INDICATED NUMBER OF JOINTS

minimize the effects of initial disturbances. However, this represents an unstable condition and is seldom encountered in common practice and, for all practical purposes, turbulent flow generally exists for a Reynolds Number above 2000 to 3000. Air flowing under all practical conditions gives Reynolds Numbers above this critical value, indicating that the flow is always turbulent. To illustrate, standard air flowing in an 8-in. duct gives a Reynolds Number of 2500 when the velocity is 37 fpm.

It has been amply shown that, provided all the dimensionless characteristics

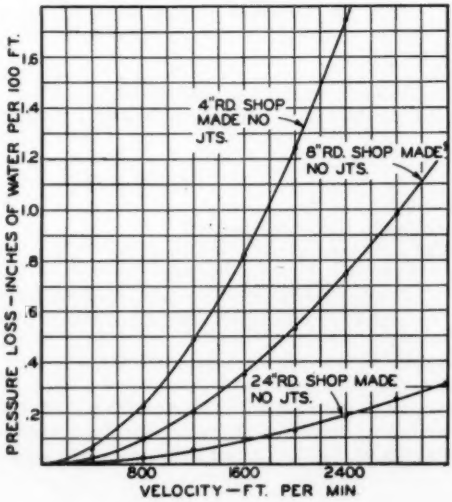

FIG. 8. RELATION BETWEEN PRESSURE LOSS AND VELOCITY FOR 24-IN., 8-IN. AND 4-IN. ROUND DUCT WITH NO JOINTS

of a pipe of a given material are similar and proportional, regardless of its diameter a single curve will express the relation between the friction factor and the Reynolds Number. The friction factor for this purpose is expressed by some authors as the reciprocal of the number of diameters length of the pipe necessary to cause a loss of one velocity head, while other investigators are accustomed to use friction factor values one-quarter smaller. The former is used in this paper but the resultant friction loss is identical in either case,

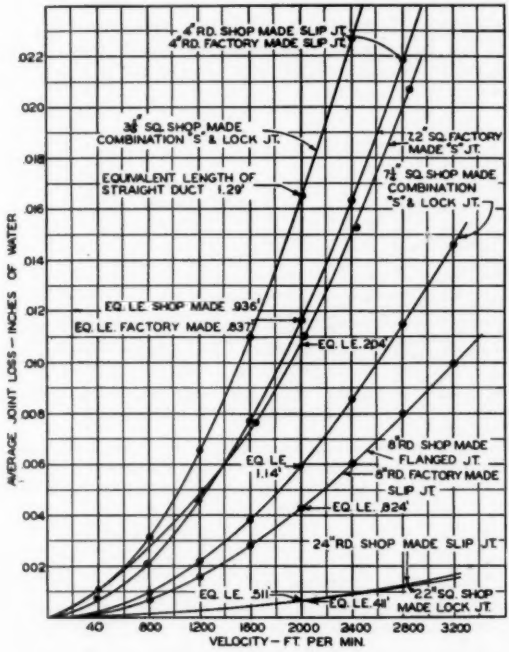

FIG. 9. RELATION BETWEEN AVERAGE JOINT LOSS AND VELOCITY FOR ALL DUCTS TESTED

since the difference in friction factor is accounted for in the respective friction loss formulae. To the extent to which this physical law is true, a single friction factor-Reynolds Number curve determined for any fluid flowing in any pipe should amply express the relationship for any other fluid flowing in this or any other pipe of the same type and material, regardless of size. The practical application of this theory in building up friction flow relationships for the flow of different liquids and gases in different types of pipe for practical use has caused a great deal of misunderstanding. One simple requirement for the application of the theory is usually forgotten, namely, the provision that all dimensionless characteristics of the different sizes of pipe of the same material

must be proportional; that is, as the diameter of the pipe changes, the roughness of the interior surface, the irregularities caused by joints, and presumably the length of sections between joints must all change in proportion to the diameter. In practice this requirement obviously never exists, and particularly is this true in the case of galvanized ducts of different sizes. The roughness of the interior of such a duct and the constriction and expansion effects at joints are practically constant regardless of the size of duct and, as a matter of fact, these irregularities may more often be greater rather than smaller in proportion to the diameter in small sizes of pipe. This is more particularly true in standard pipe. Certainly, however, in galvanized ducts the surface irregularities do not increase in proportion to the diameter; hence, a strict application of Reynolds Number should not be expected to apply.

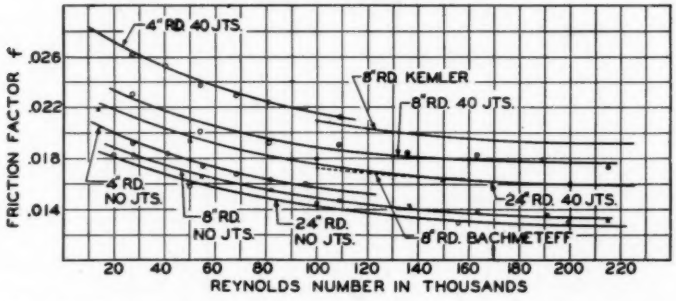

FIG. 10. RELATION BETWEEN THE FRICTION FACTOR AND REYNOLDS NUMBER FOR THE ROUND DUCTS TESTED WITHOUT JOINTS AND WITH 40 JOINTS PER 100-FT LENGTH OF RUN. CURVES FOR DATA PUBLISHED BY KEMLER [2] AND BACHMETEFF [3] ARE INCLUDED

In Fig. 10 the relation between the friction factor and the Reynolds Number, expressed in thousands, is plotted for the different sizes of round duct tested without joints and also with 40 joints per 100-ft length of run. It will be noted that for both the ducts without joints and with joints there is a consistent series of curves showing higher friction factors for the same Reynolds Number with decrease in diameter. This is as should be expected from a strict application of the Reynolds Number theory. Since the larger ducts have constrictions due to roughness, smaller in proportion to their diameter than is the case with the smaller duct, they should offer a lower friction factor index for the same Reynolds Number. It is also of interest to note the marked effect which joints in the duct have on these relationships.

For comparison, friction factor-Reynolds Number relationship curves are also included from the published results by Kemler [2] and by Bachmeteff.[3] The lack of agreement between the results of this study and the work of these investigators was the cause of a great deal of concern and extensive investigation during the early part of this study, until the importance of the proper

[2, 3] See Bibliography (2) and (3).

proportion between diameter and other dimensions including roughness was taken into consideration.

The relation between the pressure loss and volume of air flowing for the three sizes of round duct without joints is plotted in Fig. 11 for the Laboratory data. Curves based upon the Friction Chart in the ASHVE HEATING, VENTILATING, AIR CONDITIONING GUIDE, 1938 (page 566), are also included. In order to show agreement between test data for any size of duct and the

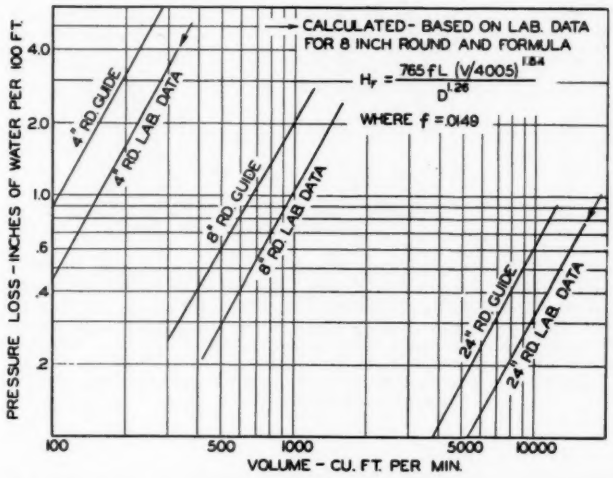

FIG. 11. RELATION BETWEEN PRESSURE LOSS AND VOLUME FOR 1938 GUIDE DATA AND THE RESULTS OF THIS STUDY WITH NO JOINTS. ARROWS INDICATE THE LOCATION OF CURVES FOR 4-IN. AND 24-IN. ROUND DUCT BASED ON CALCULATION USING 8-IN. ROUND LABORATORY DATA AND THE FORMULA GIVEN

calculated relationship based upon the modified Fanning formula, the predicted position of the curves for the 4- and 24-in. ducts, based upon the Laboratory data for the 8-in. duct and the formula,

$$H_F = \frac{0.765\, f\, L\, (V/4005)^{1.84}}{D^{1.26}} \qquad (8)$$

are indicated by arrows. It will be seen that a perfect check between the Laboratory data curves and the predicted location of the curves based upon calculation from the 8-in. curve results.

Similar relationships to those given in Fig. 11 are given in Fig. 12 for the three sizes of round duct with 40 average joints per 100-ft length of run. The

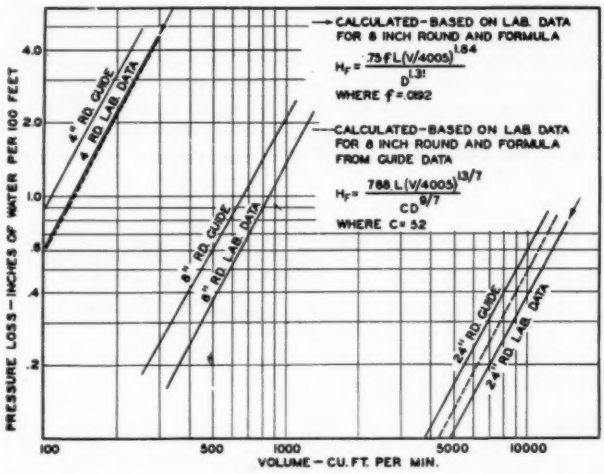

FIG. 12. RELATION BETWEEN PRESSURE LOSS AND VOLUME FOR 1938 GUIDE DATA AND THE RESULTS OF THIS STUDY WITH 40 JOINTS PER 100-FT LENGTH OF RUN. ARROWS INDICATE THE LOCATION OF CURVES FOR 4-IN. AND 24-IN. ROUND DUCT BASED ON CALCULATION USING 8-IN. ROUND LABORATORY DATA AND THE FORMULA GIVEN. BROKEN LINE CURVES INDICATE THE LOCATION OF 4-IN. AND 24-IN. ROUND LABORATORY DATA CURVES RESULTING FROM FORMULA GIVEN WHICH APPLIES TO PRESENT GUIDE DATA

arrows again indicate a perfect check between the curves based upon Laboratory tests on the 4- and 24-in. ducts and the calculated location based upon the 8-in. duct curve, and the formula:

$$H_F = \frac{0.75 f L (V/4005)^{1.84}}{D^{1.31}} \quad \ldots\ldots (9)$$

It is of interest to point out that through the choice of proper exponents and constants in the formula, it would be expected that the calculated location of the arrow would agree with the curve based on test data for at least one of the 4- or the 24-in. duct. However, the fact that the calculation applies so accurately to both the smaller and the larger duct can only be accounted for by the application of a definite mathematical relationship for the different sizes of duct. This double agreement based upon calculation by a single formula should serve to lend confidence to the acceptance of the data on the three sizes of duct as a basis for expansion through this formula to smaller, intermediate and larger ducts. Certainly when this perfect agreement applies upon extrapolating from an 8-in. to a 24-in. duct, similar extrapolation up to ducts 4 or even 6 ft in diameter should be acceptable.

The broken line curves give the location of calculated curves for the 4- and

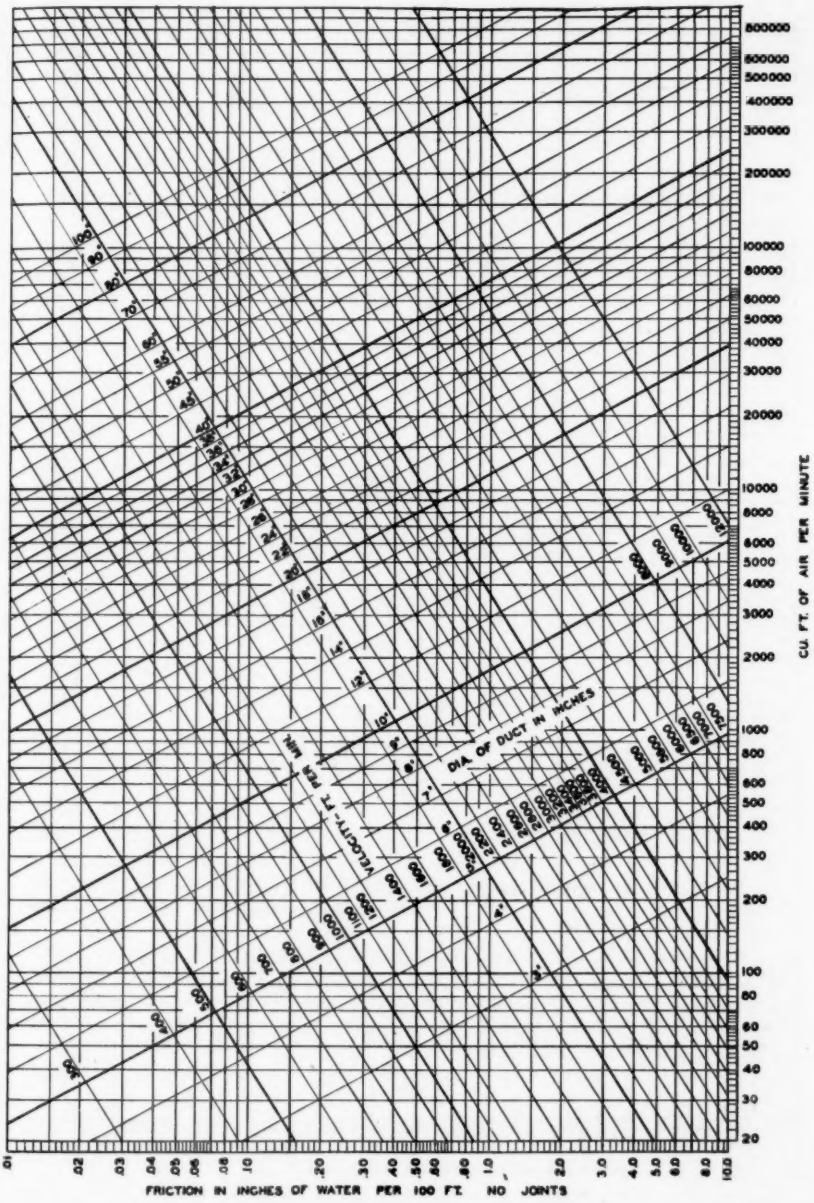

FIG. 13. FRICTION CHART BASED ON LABORATORY DATA FOR STRAIGHT RUN OF ROUND DUCT WITHOUT JOINTS. NO FACTOR OF SAFETY ALLOWED

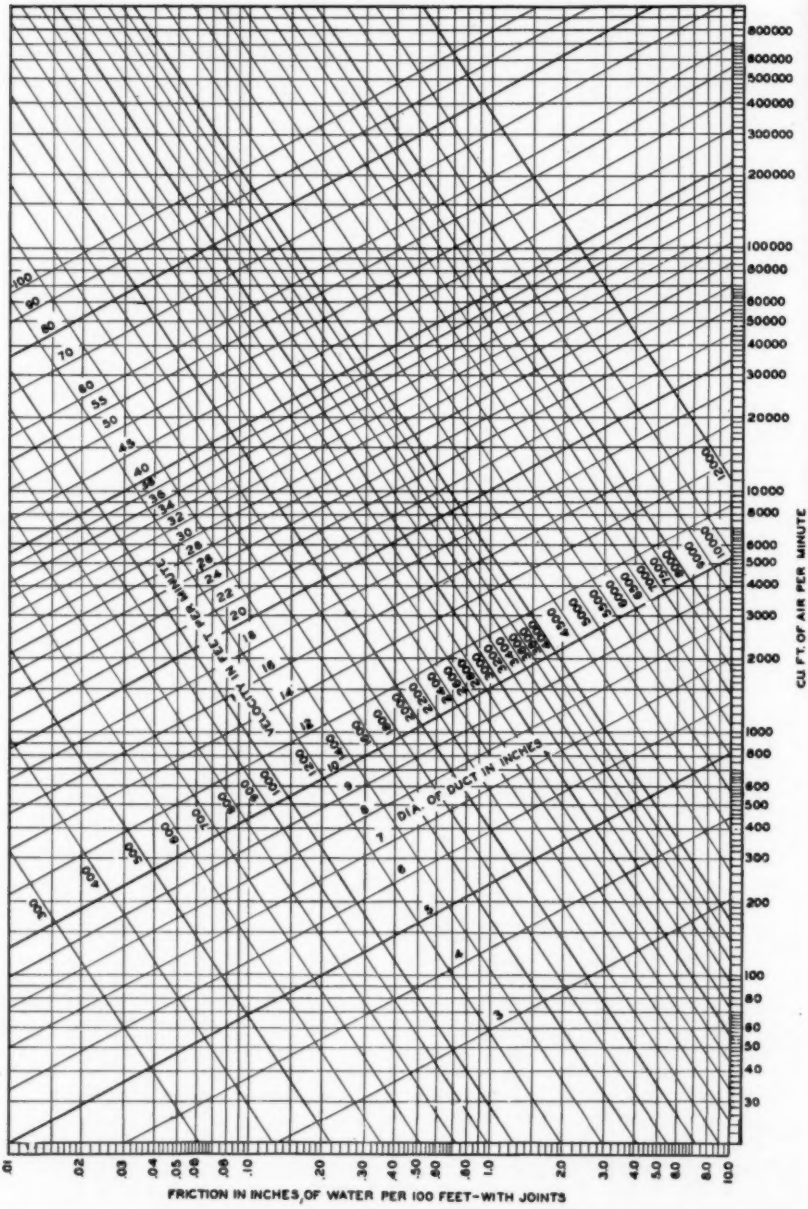
CU. FT. OF AIR PER MINUTE
DIA. OF DUCT IN INCHES
VELOCITY IN FEET PER MINUTE
FRICTION IN INCHES OF WATER PER 100 FEET—WITH JOINTS

FIG. 14. FRICTION CHART BASED ON LABORATORY DATA FOR STRAIGHT RUN OF ROUND DUCT WITH 40 JOINTS PER 100-FT LENGTH OF RUN. NO FACTOR OF SAFETY ALLOWED

24-in. ducts based upon the Laboratory data for the 8-in. duct and the formula derived from the GUIDE data:

$$H_F = \frac{0.768\, f\, L\, (V/4005)^{12/7}}{D^{9/7}} \qquad (10)$$

It will be seen from this that, in order to extrapolate with confidence to intermediate and larger sizes of duct, it was not only necessary to establish an accurate curve for a given size of duct, but also to modify the constants and exponents in the formula by which the expansion was to be made.

DESIGN DATA

Based upon the formulae developed and used in connection with Figs. 11 and 12, the Laboratory data have been expanded into friction charts for a wide range of sizes of round duct without joints, and with 40 joints per 100-ft length of run, in Figs. 13 and 14, respectively. These charts are recommended for practical use. As pointed out a careful study of the relation between the number of joints per length of run as found in actual systems and as recommended by different authorities indicates an average arrangement of about one joint per every 2½ ft or 40 joints per 100-ft length of run. Based on the assumption that this frequency of joints will fit average design practice best, Fig. 14 is assumed to apply. However, in order to give the designing engineer an understanding of the effect of joints, as well as a basis on which a design for any other duct arrangement may be corrected, the friction chart for duct with no joints, Fig. 13, is also presented.

The original plans for the duct size study included a comprehensive study of rectangular ducts of various aspect ratios, in order to establish a basis for more accurate conversion from such rectangular ducts to equivalent round. The limitation of the study to square ducts, as carried out to date, with the one exception of rectangular duct, makes the development of a conversion relationship a little uncertain. However, based upon this limited study conversion data are offered in Table 2. The data are applicable with sufficient accuracy to convert from the square to round, and may, therefore, be assumed to apply without great error for other rectangular ducts. This phase of the subject should be investigated further.

Table 2 is a conversion chart giving the equivalent round pipe (same friction loss for same volume) for various rectangular ducts. For rectangular ducts it has been found that joints occur approximately every 5 ft, and for this reason the conversion chart is based on changing rectangular duct with 20 joints per 100 ft of run to round duct with 40 joints per 100 ft. A convenient formula, using the derived friction loss formula and the hydraulic diameter relationship, $D = 4A/P$,

where:

D = diameter in inches
A = cross-sectional area in square inches
P = wetted perimeter in inches,

TABLE 2—CIRCULAR EQUIVALENTS OF RECTANGULAR DUCTS FOR EQUAL FRICTION AND CAPACITY

SIDE RECTANGULAR DUCT—INCHES	3	3¼	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	22	24
6	4.7	4.9	5.5	6.1	6.7	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
7	5.0	5.2	6.0	6.6	7.3	8.0	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
8	5.4	5.7	6.3	7.1	7.9	8.5	9.1	...	...	...	...	...	...	...	...	...	...	...	...	...	...
9	5.7	6.0	6.7	7.5	8.3	9.0	9.6	10.2	...	...	...	...	...	...	...	...	...	...	...	...	...
10	5.8	6.2	7.0	7.9	8.7	9.5	10.1	10.8	11.4	...	...	...	...	...	...	...	...	...	...	...	...
11	6.2	6.5	7.3	8.3	9.1	9.9	10.6	11.2	11.9	12.5	...	...	...	...	...	...	...	...	...	...	...
12	6.4	6.8	7.6	8.6	9.5	10.3	11.1	11.8	12.5	13.0	13.6	...	...	...	...	...	...	...	...	...	...
13	6.7	7.0	7.8	8.9	9.9	10.7	11.5	12.2	13.0	13.6	14.2	14.8	...	...	...	...	...	...	...	...	...
14	6.9	7.2	8.1	9.2	10.2	11.1	12.0	12.7	13.4	14.1	14.8	15.4	16.0	...	...	...	...	...	...	...	...
15	7.1	7.4	8.4	9.5	10.5	11.4	12.3	13.1	13.9	14.5	15.2	15.9	16.5	17.1	...	...	...	...	...	...	...
16	7.4	7.6	8.6	9.7	10.8	11.8	12.7	13.5	14.3	15.0	15.7	16.4	17.1	17.7	18.1	...	...	...	...	...	...
17	7.5	7.8	8.8	10.0	11.1	12.1	13.1	13.9	14.7	15.5	16.3	17.0	17.6	18.0	18.8	19.3	...	...	...	...	...
18	7.7	8.0	9.0	10.2	11.5	12.5	13.5	14.3	15.1	15.9	16.7	17.4	18.1	18.8	19.4	20.0	20.5	...	...	...	...
19	7.9	8.1	9.2	10.5	11.7	12.7	13.7	14.6	15.5	16.3	17.1	17.9	18.5	19.2	19.8	20.3	20.8	21.2	...	...	...
20	8.0	8.3	9.4	10.7	12.0	13.0	14.0	15.0	15.9	16.7	17.5	18.3	19.0	19.7	20.4	21.0	21.6	22.1	22.7	...	...
22	8.3	8.7	9.7	11.1	12.4	13.4	14.6	15.6	16.6	17.5	18.3	19.1	19.9	20.6	21.3	22.0	22.7	23.3	23.8	24.9	...
24	8.6	9.0	10.1	11.5	12.9	14.1	15.2	16.2	17.2	18.1	18.9	19.8	20.5	21.4	22.1	22.9	23.6	24.3	25.0	26.2	27.3
26	8.9	9.3	10.4	11.9	13.3	14.6	15.6	16.8	17.9	18.8	19.8	20.7	21.5	22.3	23.1	23.9	24.6	25.3	25.9	27.2	28.4
28	9.2	9.5	10.8	12.3	13.8	15.1	16.3	17.4	18.5	19.3	20.3	21.2	22.2	23.0	23.8	24.6	25.4	26.0	26.8	28.1	29.4
30	9.4	9.7	11.2	12.7	14.2	15.5	16.7	17.9	19.0	20.0	20.9	21.9	22.8	23.7	24.5	25.4	26.1	26.9	27.7	29.2	30.5
32	9.7	10.1	11.5	13.1	14.6	15.9	17.1	18.3	19.6	20.7	21.6	22.5	23.5	24.4	25.3	26.2	27.0	27.9	28.6	30.1	31.6
34	10.0	10.3	11.8	13.2	14.9	16.3	17.7	18.9	20.0	21.1	22.1	23.1	24.0	25.0	25.9	26.8	27.6	28.5	29.4	30.9	32.4
36	10.1	10.6	12.0	13.7	15.3	16.8	18.0	19.4	20.6	21.8	22.8	24.0	25.0	26.0	27.0	27.9	28.7	29.6	30.3	32.0	33.5
38	10.3	10.8	12.3	14.0	15.5	17.0	18.5	19.8	21.0	22.1	23.3	24.4	25.4	26.4	27.5	28.4	29.4	30.3	31.1	32.7	34.2
40	10.5	11.0	12.5	14.1	15.9	17.5	18.9	20.2	21.6	22.8	24.0	25.1	26.2	27.2	28.1	29.1	30.3	30.9	31.8	33.4	34.9
42	10.7	11.3	12.8	14.5	16.3	17.7	19.3	20.6	22.0	23.3	24.4	25.5	26.6	27.7	28.7	29.7	30.7	31.6	32.5	34.2	35.9
44	10.9	11.4	13.0	14.8	16.5	18.1	19.7	21.1	22.4	23.6	24.9	26.0	27.1	28.2	29.3	30.2	31.2	32.1	33.0	34.8	36.5
46	11.1	11.6	13.2	15.1	16.8	18.5	19.9	21.5	22.8	24.2	25.4	26.7	27.9	29.0	30.1	31.2	32.3	33.1	34.0	35.8	37.4
48	11.2	11.8	13.4	15.3	17.1	18.8	20.4	21.8	23.2	24.5	25.9	27.1	28.3	29.4	30.6	31.7	32.7	33.7	34.6	36.4	38.1
50	11.5	12.0	13.6	15.6	17.5	19.2	20.7	22.2	23.6	25.0	26.3	27.6	28.9	30.0	31.1	32.3	33.3	34.3	35.2	37.0	38.7
52	11.6	12.2	13.8	15.8	17.8	19.5	21.0	22.6	23.9	25.5	26.7	28.0	29.2	30.5	31.6	32.8	33.9	34.8	35.9	37.8	39.7
54	11.8	12.4	14.0	16.1	18.0	19.7	21.4	22.9	24.3	25.8	27.3	28.5	29.7	30.9	32.1	33.2	34.2	35.3	36.3	38.3	40.3
56	12.0	12.6	14.3	16.3	18.2	20.0	21.6	23.4	24.9	26.2	27.6	29.0	30.3	31.6	32.9	34.0	35.1	36.2	37.2	39.2	41.1
58	12.1	12.7	14.4	16.6	18.4	20.1	22.0	23.6	25.2	26.4	28.1	29.5	30.9	32.1	33.3	34.5	35.6	36.7	37.5	39.7	41.6
60	12.3	12.9	14.6	16.8	18.7	20.3	22.4	23.8	25.5	27.0	28.6	30.0	31.4	32.6	33.9	35.1	36.2	37.4	38.4	40.6	42.6
62	12.4	13.0	14.9	17.2	19.2	20.8	22.8	24.4	25.9	27.3	28.8	30.2	31.5	32.8	34.0	35.2	36.3	37.5	38.5	40.8	42.9
64	12.5	13.1	15.1	17.3	19.3	21.2	22.9	24.6	26.2	27.7	29.1	30.5	31.8	33.1	34.3	35.5	36.7	37.8	38.9	41.0	43.1
66	12.7	13.4	15.3	17.6	19.6	21.5	23.3	25.0	26.6	28.1	29.6	30.9	32.3	33.6	34.8	36.1	37.3	38.4	39.6	41.9	43.9

TABLE 2—(*Continued*)

SIDE RECTANGULAR DUCT—INCHES	26	28	30	32	34	36	38	40	42	44	46	48
26	29.5	...	...	...	...	...	...	...	...	...	...	...
28	30.5	31.6	...	...	...	...	...	...	...	...	...	...
30	31.9	33.0	34.2	...	...	...	...	...	...	...	...	...
32	33.0	34.4	35.7	36.9	...	...	...	...	...	...	...	...
34	33.9	35.2	36.5	37.7	38.7	...	...	...	...	...	...	...
36	34.9	36.3	37.6	38.8	39.9	41.0	...	...	...	...	...	...
38	35.7	37.1	38.5	39.9	41.1	42.3	43.5	...	...	...	...	...
40	36.4	37.9	39.3	40.7	42.0	43.2	44.4	45.5	...	...	...	...
42	37.5	39.0	40.5	41.8	43.1	44.4	45.6	46.8	47.9	...	...	...
44	38.2	39.9	41.4	42.9	44.2	45.6	46.8	48.0	49.1	50.1	...	...
46	39.1	40.7	42.2	43.8	45.1	46.4	47.7	49.0	50.1	51.2	52.1	...
48	39.8	41.4	42.9	44.0	45.9	47.3	48.7	50.0	51.4	52.6	53.8	54.9
50	40.4	42.1	43.7	45.3	46.8	48.2	49.7	51.0	52.4	53.6	54.7	55.7
52	41.2	42.8	44.5	46.2	47.5	48.9	50.4	51.7	53.0	54.9	56.1	57.2
54	42.1	43.8	45.5	46.9	48.4	49.9	51.3	52.7	54.0	55.4	56.7	58.0
56	42.9	44.5	46.0	47.7	49.2	50.8	52.2	53.7	55.0	56.4	57.6	58.9
58	43.4	45.0	46.7	48.6	50.1	51.5	53.0	54.5	55.8	57.3	58.5	59.8
60	44.0	45.7	47.4	49.2	50.7	52.5	53.9	55.4	56.6	58.2	59.7	61.1
62	44.5	46.4	48.2	50.0	51.7	53.7	54.7	56.2	57.7	59.1	60.5	61.9
64	45.0	46.9	48.8	50.5	52.3	53.9	55.6	57.2	58.7	60.1	61.6	62.8
66	45.9	47.8	49.6	51.3	53.0	54.6	56.2	57.8	59.4	60.9	62.3	63.8
68	46.6	48.5	50.3	52.1	53.9	55.5	57.1	58.8	60.3	61.9	63.3	64.7
70	46.9	48.9	50.8	52.6	54.4	56.0	57.8	59.6	61.1	62.7	64.2	65.7
72	47.7	49.6	51.6	53.3	55.1	56.8	58.5	60.2	61.8	63.4	64.9	66.4

SIDE RECTANGULAR DUCT—INCHES	50	54	60	66	72	78	84	88
50	56.7	...	...	...	...	...	...	...
52	58.1	...	...	...	...	...	...	...
54	59.3	61.7	...	...	...	...	...	...
56	60.0	62.4	...	...	...	...	...	...
58	61.9	64.1	...	...	...	...	...	...
60	62.4	65.0	68.2	...	...	...	...	...
62	63.2	65.5	69.4	...	...	...	...	...
64	64.0	66.8	70.6	...	...	...	...	...
66	65.2	67.9	71.5	75.1	...	...	...	...
68	66.1	68.8	72.6	76.7	...	...	...	...
70	67.1	69.9	73.6	77.6	...	...	...	...
72	67.8	70.6	74.5	78.6	81.8	...	...	...
74	68.8	71.5	75.6	80.2	83.3	...	...	...
76	69.8	72.6	76.8	80.6	84.3	...	...	...
78	70.4	73.4	77.7	81.7	85.3	88.5	...	...
80	71.3	74.4	78.7	82.6	86.2	89.7	...	...
82	72.2	75.3	79.6	83.5	87.2	90.8	...	...
84	73.0	75.8	80.1	84.3	88.4	92.1	95.5	...
86	73.7	76.8	81.3	85.6	89.5	93.3	96.8	...
88	74.6	77.8	82.3	86.6	90.5	94.3	97.9	100.2
90	75.3	78.3	82.9	87.2	91.3	95.1	98.9	101.2
92	76.2	79.4	84.0	88.4	92.4	96.4	100.1	103.0
94	77.1	80.2	84.8	89.1	93.3	97.3	101.1	103.5
96	77.7	80.9	85.7	90.1	94.3	98.4	102.2	104.5

is as follows:

$$D = 1.306 \sqrt[4.94]{\frac{(a\,b)^{3.10}}{(a+b)^{1.36}}} \quad \ldots\ldots(11)$$

where:

a = width of duct
b = height of duct.

Using this formula, the equivalent square duct for 4-in. round, 8-in. round and 24-in. round, was calculated. This formula was found to be slightly in error when results from its application were compared with tests on the round and square ducts. The formula was therefore modified by changing the constant to give:

$$D = 1.32 \sqrt[4.94]{\frac{(a\,b)^{3.10}}{(a+b)^{1.36}}} \quad \ldots\ldots(12)$$

which holds true for ducts with no joints or equal number of joints. To convert rectangular ducts with 20 average joints per 100 ft of run to round duct with 40 average joints, the modified formula becomes

$$D = 1.35 \sqrt[4.94]{\frac{(a\,b)^{3.10}}{(a+b)^{1.36}}} \quad \ldots\ldots(13)$$

This formula was used in developing Table 2 which is to be used in conjunction with Fig. 13. For example, assume that the friction loss for a 16 × 14-in. duct at 2000 fpm air velocity is required. Referring to Table 2, a 16 × 14-in. duct is equivalent to a 17.1-in. round duct; referring to Fig. 13, at 2000 fpm in a 17.1-in. round duct, the loss is approximately 0.255 in. of water per 100 ft.

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DISCUSSION

F. R. BICHOWSKY: The modified Fanning formula, which was used for comparison, is, as I understand it, an empirical formula. The question that I wanted to ask was how far the constants in that formula depended on this experimental work and how far they were dependent on other factors.

F. C. HOUGHTEN: The changes in the constants in the Fanning formula were based entirely on the data presented in the paper. I do not remember just the percentage magnitude, but they are given in the paper. As brought out by Mr. Bichowsky, the Fanning formula is empirical. The constants in the formula were changed to make the data found for the 8-in. duct extrapolate properly to fit the 4- and 24-in. ducts.

No. 1106

AIR CONDITIONING IN INDUSTRY

Physiological Reactions of Individual Workers to High Effective Temperatures

By W. L. FLEISHER,* A. E. STACEY, JR.,** F. C. HOUGHTEN,*** AND M. B. FERDERBER, M.D.,† (*MEMBERS*)

THE work of the ASHVE Research Laboratory during the past summer was limited to a study of the physiological reactions of men while carrying on light work in atmospheric conditions including effective temperatures from 77 to 92 deg, with relative humidities of 60, 75 and 90 per cent. The investigations were conducted in the psychrometric chambers of the Laboratory located in the Bureau of Mines Building in Pittsburgh, under the following Technical Advisory Committee on Air Conditioning in Industry: A. E. Stacey, Jr., *Chairman;* Philip Drinker, Dr. Leonard Greenburg, H. P. Greenwald, A. M. Kinney, J. W. Kreuttner, L. L. Lewis, Dr. W. J. McConnell, Dr. C. P. McCord, P. A. McKittrick, and Dr. R. R. Sayers.

INTRODUCTION

It is well in beginning a paper of this nature to outline briefly the history of previous investigations and list the several factors which are generic to the subject. Eleven basic points are enumerated herewith which have an important relationship to proper air conditioning in industry.

1. The simultaneous relationship of temperature and humidity to the reactions of human beings *at work.*
2. The air conditions required for the making of goods, and the air conditions encountered in the extraction of basic materials.
3. The change in the required atmospheric conditions with changes in the man-power output.
4. The increased amount of work healthfully obtained with better conditions.

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* Chairman, Committee on Research, New York, N. Y.
** Chairman, Research Technical Advisory Committee on Air Conditioning in Industry, New York, N. Y.
*** Director, ASHVE Research Laboratory, Pittsburgh, Pa.
† Fellow in Fever Therapy, Dept. Industrial Hygiene, University of Pittsburgh and Advisor on Physiological Studies, ASHVE Research Laboratory, Pittsburgh, Pa.

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5. The problem of dust, vapors, fumes and gases which may be harmful to workers.
6. The development of bacteria, spores, etc., that may produce unsanitary conditions or be harmful to processes.
7. The prevalence of odors arising from working conditions.
8. The prevalence of noise disturbances.
9. Legislation required to protect labor and capital.
10. The engineering basis on which the requirements called for by legislation or determined by research can be calculated.
11. Analysis of research already done or being done or that research required to be done on which to base sound conclusions.

Underground mining, which eventually brought about research covering the requirement of human beings for proper air conditions in industry, was for thousands of years the most discredited of all industrial work. A freeman would not work underground in close and confined spaces, so the laborious, dark, wet and dangerous work of extracting minerals from underground was allotted to slaves, criminals and children. Due to lack of tools and technology, the difficulty of piercing the earth to explore and exploit the veins was so exhausting, that only the smallest amount of materials was removed. Children of from six to twelve years of age dragged sacks weighing as much as 45 lb through minute passages for hours at a time. Coal and tin, copper and gold and silver were the prizes. If it were not for the fact that the world at present is suffering from a retrogression of humanitarian ideas, approaching a general undermining of all the progress achieved in 2500 years, one might dwell longer on the industrial use of children in the production of wealth.

Early mines uncovered in archeological searches have disclosed the skeletons of hundreds of workers who undoubtedly died from the heat and dampness, combined with the efforts that they were compelled to exert. It is obvious that under such callous conditions, no thought was given to the health and safety of the worker.

Egyptian, Phoenician, Cretan, Greek, Roman and Spanish records all depict deep mining from at least 5000 B.C. Progressing down to present times and particularly in the light of recent exploitation of deep mine ventilation in South Africa, the following quotation from the 11th Edition of *Encyclopaedia Brittanica*, published in 1910, Vol. XVIII, under Deep Mining is pertinent:

"The deepest mine in the world is the No. 3 shaft of the Tamarach Mine in Houghton County, Michigan, which has reached a vertical depth of about 5200 feet.....

In the Transvaal gold region (South Africa), a number of shafts have been sunk to strike the reef at about 4000 feet. In most cases the deposits worked are known to extend to much greater depths than have been reached. The possibility of pumping and hoisting from great depths has been discussed, and it remains now to consider the other conditions which will tend to limit other mining operations in depth; namely, increase of temperature and increase of rock pressure. Observations in different parts of the world have shown that the increase of temperature in depth varies, in most localities the rise being at the rate of 1 deg for 50 to 100 feet of depth, while in the deep mines of Michigan and the Rand an increase as low as 1 deg for each 200 feet has been observed. In the Comstock Mines at Virginia City, Nevada, it is possible to continue mining operation at rock temperatures of 130 degrees. In these mines a constant supply of pure air, about 1000 cubic feet per minute, was blown into the hot working places through light iron pipes. The air issuing from these pipes was dry and warm and served to keep the temperature of the air below 120 degrees, at which temperature it was possible for men to work continuously for ½ hour at a time and for 4 hours in a day. In some places work was conducted

with rock temperatures as high as 150 deg F with air 135 deg. In these very hot drifts the fatality was large. In the Alpine tunnels, where the air was *moist* and probably not as pure as in the Comstock Mines, great difficulties were experienced in prosecuting the work at temperature of 90 deg F or less. The mortality was large and it was believed by the engineers that temperatures over 104 deg would have proved fatal to most of the workmen. Deep mines, however, are generally dry so that in most cases it will be possible to realize the more favorable conditions of the Comstock Mines. Assuming an initial mean temperature of 50 deg F and increments of 1 deg for 100 and for 200 feet, a rock temperature of 130 deg will be reached at 8,000 to 16,000 feet."

This quotation is reproduced in order to give evidence that the rise in temperature with depth of mine was early understood and resultant conditions for greater depths prophesied, even though the depths mentioned had not then been reached. It is also evident from this statement that moisture added to the sensible heat was understood to have a deleterious effect on the physiological reactions of workers.

Deep mining has been used as an illustration because it existed long before machine-operated industrial activity was prevalent. In fact, the establishment of the ASHVE Research Laboratory at Pittsburgh revolves about the interest of the U. S. Bureau of Mines in the health and working conditions of labor in mining.

The history of the legislation and investigation of ventilation, particularly in coal mines, dates back only about a hundred years. In England, an investigation of the reasons for explosions in coal mines was started in 1835 by an Act of Parliament, when a committee was appointed to investigate "lamentable catastrophes which have occurred in the mines of Great Britain, with the view of ascertaining and suggesting means for preventing the recurrence of similar fatal accidents." The committee reported to Parliament, but no laws were enacted although the investigation brought out facts and recommendations.

In 1843 a law making the inspection of mines a state function was passed by Parliament. This appears to be the first law, either in the United States or England, where state regulation of mining was enacted. Subsequently great interest was taken in solving the ventilation problems in Great Britain. Although certain methods were recommended the mine owners neglected to apply the suggested methods because of lacking compulsory requirements, until a continuous list of disasters brought about a new Act of Parliament in 1850, placing all coal mines under state inspection.

The United States was slower in considering this question, and although the miners petitioned the Pennsylvania Legislature in 1858 for the same type of laws as those governing the mines in Great Britain, no action was taken on their petition until an explosion in 1869 at Avondale, Pennsylvania, forcibly brought the matter of mine ventilation to the attention of the public. This disaster was responsible for an Act of the Pennsylvania Legislature, in 1870, which stated:

"In a non-gaseous mine, the minimum quantity of air shall not be less than 150 cubic feet per minute for each person employed. In a mine wherein explosive gas is being generated . . . the minimum quantity of air shall not be less than 200 cubic feet of air per minute for each person employed therein, and as much more in either case as one or more of the inspectors may deem requisite."

The historical background of ventilation in mining has been developed so that a differentiation may be given between present research activities and

those researches that had to do with the conditions existing in mines. It was obvious for centuries that something had to be done in the deep, dark holes in

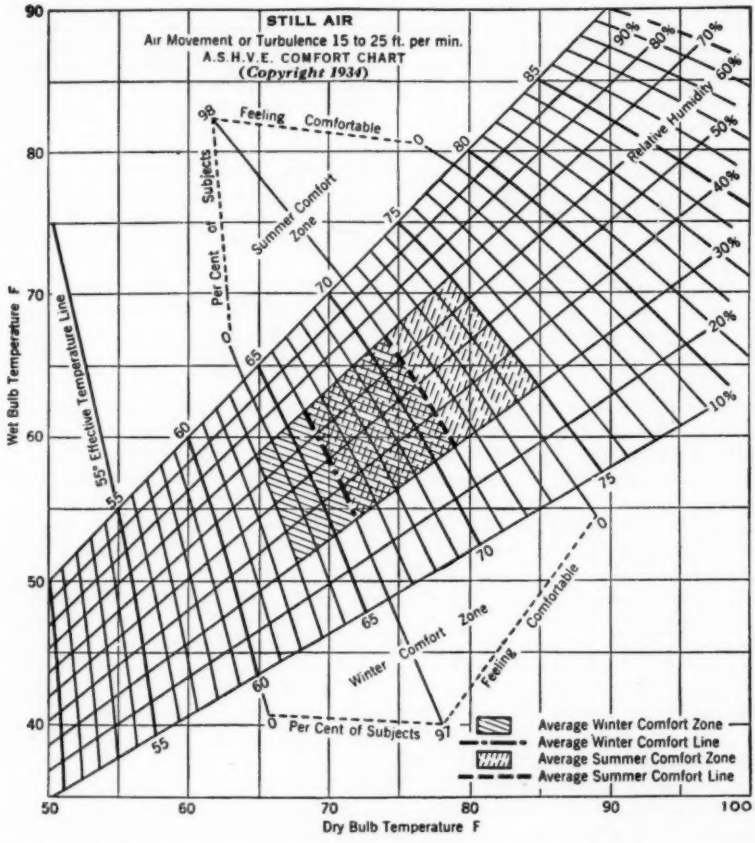

FIG. 1. ASHVE COMFORT CHART FOR AIR VELOCITIES OF 15 TO 25 FPM (STILL AIR)

Note.—Both summer and winter comfort zones apply to inhabitants of the United States only. Application of winter comfort line is further limited to rooms heated by central station systems of the convection type. The line does not apply to rooms heated by radiant methods. Application of summer comfort line is limited to homes, offices and the like, where the occupants become fully adapted to the artificial air conditions. The line does not apply to theaters, department stores, and the like where the exposure is less than 3 hours.

which mining was conducted and in which either explosive gases or dusts were so prevalent. However, it was really the requirements of such industries as the textile industries which prompted the present investigations. It has

become evident by gradual education that the power required to manufacture products at high speeds, such as are now existent in the textile plants, has created enormous amounts of heat by friction and that all of the power developed in industrial plants is translated into sensible heat. It is only within the last few years that engineers have realized the effect of radiant heat as producing a high resultant total heat in an enclosure. Still later it has been realized that the energy output of human beings is translated into both sensible and latent heat, which is added to these other factors, and that all of these items are cumulative. It has been further shown that hygroscopic material functions at a definite relative humidity which is independent, to a great extent, of the dry-bulb temperature. As the temperature from the sources mentioned rises inside an enclosure in which hygroscopic material is being developed and manufactured, more and more moisture must be added to the air in order to maintain a constant relative humidity. A decrease in the relative humidity produces, on high-speed machines working with dialectric materials, static electricity which makes the operation of the machines and the production of properly finished goods difficult unless the relative humidity is maintained at the proper desirable high levels. All of these factors combined tend to create conditions affecting the physiological reactions of the workers. None of the former investigations visualized the correlation of these different factors, and it is felt that although the basic ideas have been more or less prophesied to a great extent, they have been definitely overlooked and neglected, notwithstanding their great importance to both labor and capital.

No excuse is being offered for the research that has been undertaken, but it is important to bring to the attention of the public the great necessity for this correlated research as the beginning of specific research in many different industries. In order to present some of the work that has been accomplished certain facts are given which influenced the methods of investigation.

Over a period of many years research has been directed in an effort to determine a zone of comfort. In this research it was discovered that within certain wide limits, the individual experienced equal comfort at widely different points on the dry-bulb scale with different wet-bulb or relative humidity relationships as illustrated in Fig. 1. In summer equal comfort was experienced by large groups with maintained conditions of 76 F dry-bulb and 75 per cent relative humidity, and 78 F dry-bulb and 50 per cent relative humidity, and 82 F dry-bulb and 30 per cent relative humidity. All of these results were obtained with air velocities of 15 to 25 fpm. The line drawn through these points was designated as an effective temperature line and designated by the temperature of saturation intersected by the line or curve through these three points as a 73 deg ET line.

These investigations covering a period of 15 years are still being conducted with astonishingly accurate results, although many questionable points are still to be determined. Practically all of these investigations have been, at least lately, concerned with cooling—with people at rest—and with the questions of shock, drafts, removal of perspiration and the cycle of sensations: cold, warm, hot, comfortable. The object of the present research is undertaken to proceed deeper and further than comfort research, because no attempts are being made to indicate the results by physical reactions, or sensations, but primarily to secure the basic physiological data.

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Similar investigations in this field were begun about 1860. In the report of the New York State Commission on Ventilation, a very comprehensive historical account of the investigations of earlier medical workers is given in a most interesting review. A few of the points from that account are presented herewith to indicate a distinct trend towards the investigations referred to in this discussion. Cardinal Polignac, in 1713, stated that inequalities of temperature and want of ventilation were responsible for numerous maladies. In 1883, F. T. F. Hennans, in *Archives für Hygiene,* discussed the strong probabilities that the influences exerted by the air of a badly ventilated room were thermal. He gave credit for this theory to Pettenkover. It was only in 1905 that Flügge established the thermal theory of ventilation on a fine basis. From that time on J. S. Haldane and his associates very definitely enunciated the theory of the deleterious effects of high temperatures, combined with high wet-bulbs. A paragraph is quoted from the Ventilation Commission Report, which bears directly on this subject:

"Evidence in favor of the views advanced by Flügge and his associates accumulated rapidly during the immediately succeeding years, particularly in England. Striking evidence in regard to the effect of high temperature combined with high humidity had been obtained by Haldane, Pembrey, Boycott and Hill in the course of studies on problems of industrial hygiene. In the year that Flügge's researches were published Haldane reported that the rectal body temperature begins to rise when the wet bulb temperature of the air reaches 34.4 degrees C. (93 F.) if the subject be at rest in moving air, and that the same condition appears at 31 deg. C. (88 F.) wet bulb if the subject be at rest in still air, at 30½ deg. C. (88 F.) wet bulb when exercising in moving air and at 25½ deg. C. (78 F.) when exercising in still air. A marked increased in pulse rate accompanied by throbbing in the head, dysnia and a general feeling of exhaustion and discomfort accompany the rise in rectal temperature."

Hill, Flack, MacIntosh, Rollins and Walker, writing of *The Influence of the Atmosphere to Our Health and Comfort in Confined and Crowded Spaces* (*Smithsonian Miscellaneous Collections,* Vol. 60, No. 23, Publication 2170) reported:

"In one class of experiments we shut within the chamber seven or eight students for about a half an hour and observed the effect of the combined atmosphere upon them. . . . The wet bulb temperature rose meanwhile to about 26.6 to 29½ C. (80 to 85 F.) and the dry bulb a degree or two higher. The students went on chatting and laughing but bye and bye as the temperature rose they ceased to talk and their faces became flushed and moist. . . . Their discomfort was relieved to an astonishing extent by putting on the electric fans placed on the roof."

The investigations of the New York State Commission, initiated in 1913, covered almost every phase of human reaction to temperature and humidities and the consequent wet-bulbs, within the average ranges likely to be experienced in school-houses, from the most favorable to the worst conditions correlating these conditions with different physical and mental endeavors which might be exerted by the students.

Society research, undertaken at the Laboratory in Pittsburgh, starting in 1920, indicated that the upper limit of the effective temperature to which the human organism is capable of adapting itself without serious discomfort or injury to health is 90 deg ET for men at rest and between 80 and 90 deg ET for men at work, depending upon the rate of work. An effective temperature of 85 deg may mean 87 F dry-bulb and 84 F wet-bulb or 100 F dry-bulb and

76 F wet-bulb; the first giving a relative humidity of 90 per cent and the second a relative humidity of 33 per cent. Or this same relationship might represent a dry-bulb temperature of 114 F and a wet-bulb of 65 F and about 3 per cent relative humidity, all of which may be referred to in Fig. 2.

In 1923 the Research Laboratory conducted a series of tests [1, a] which indicated that the wet-bulb temperatures above 85 F seriously upset the physiological conditions of the subjects, as shown by rise in rectal temperature, pulse rate and blood count, but with no serious change in the metabolism of the subjects. In the summary the authors state:

"1. There is an inability of the body at rest and in still air to compensate for saturated atmospheric conditions exceeding 90 F.
2. The physiological effects resulting from exposure to high temperatures and humidities depend upon both the wet- and dry-bulb readings.
3. The exhaustion and weakness following subjection of human beings to a very high temperature and humidity for a short period is not so severe as subjection to a moderately high temperature and humidity, but for a longer period.
4. The highest dry- and wet-bulb temperatures attained and length of time endured in the experiments are as follows:

DRY-BULB TEMP, F	WET-BULB TEMP, F	RELATIVE HUMIDITY, PER CENT	TIME, MINUTES
112.5	112.5	100	35
120.2	104.02	60	40
147	108.4	30	45
157	100.43	15	45

5. The pulse-rate, rather than the rise in body temperature, apparently determines the extent of the discomfort experienced by the subject. Subjects became very uncomfortable after the pulse-rate exceeded 135 pulsations per min, and complained of unbearable and distressing symptoms when the pulse exceeded 160 per min. The highest pulse rate recorded was 184 per min.
6. The systolic and diastolic blood pressure fell with moderate increase in temperature and humidity, and the systolic rose and the diastolic fell, thus increasing the pulse pressure, in high temperatures and humidities."

The reason for quoting this summary is to indicate the limits which the early tests covered. The purpose in re-establishing this most essential research was to simulate actual working conditions for a group of people, that would approach, from the standpoint of bearable effective temperatures, what might be expected, or that should be produced, so that men and women, without injury, could work for the periods of time essential in such industrial establishments under the most favorable conditions that could be economically produced. Also to determine from a sufficiently large group, the variations in the physiology of different subjects so that a method of determining those possible divergences in human beings may be established for the selection of a worker in a particular industry.

With the movement of the textile industry from New England and Pennsylvania to the south and the increasing demands of labor for better and more healthful conditions, legislation was proposed in various states to protect the

[a] Reference numbers refer to listing of papers in bibliography.

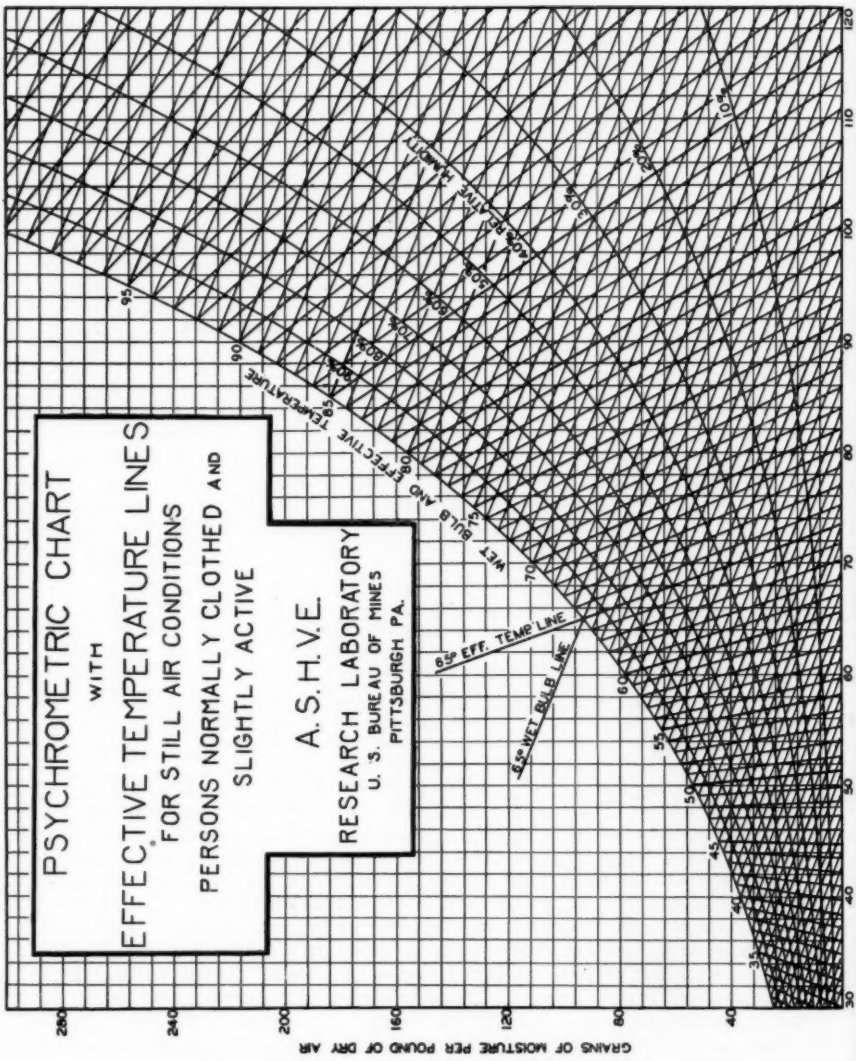

FIG. 2. PSYCHROMETRIC CHART FOR HIGH EFFECTIVE TEMPERATURES

worker. Practically all legislation called for *proper ventilation.* It was with the idea of interpreting for the world this indefinite expression concerning proper ventilation that this work was begun based on already accomplished but incomplete research, which has previously been quoted in a meager summary. It was not only the purpose of this research to determine, if possible, at what point in conditions which may readily occur in industry a physiological break takes place in the workman, but also to outline from an engineering standpoint what methods may be pursued, from an air conditioning angle, to relieve or prevent the occurrence of critical conditions.

In the textile field, particularly in the south, the average mill building, unless properly ventilated and treated either by cooled air or large volumes of circulated air, may have a temperature in the summertime as high as 120 F due to transmission, solar radiation, motor or frictional input and the metabolism of the workers. When it is realized that relative humidities of 70 per cent must be maintained in these rooms, it can be seen that unless the dry-bulb temperatures are reduced the resulting wet-bulb with 120 F and even 60 per cent relative humidity (which relative humidity gives a wet-bulb of 108 F) would be unbearable for more than a few minutes. It is therefore necessary in industrial plants of this kind to reduce the temperatures in order to maintain the relative humidities required by the products.

It is the purpose of this research to attempt to establish, by the use of the laboratory methods, a critical point at which the majority of people, working for a definite period—then a rest period, as would normally occur in industry—then a continued work period for a definite length of time, and with the work repeated from day to day, would show no deleterious effect at the end of an average working week. The whole attempt has been and will continue to be to use the laboratory as a pilot plant to fix conditions which will be developed in the field and from which eventually both the engineering of required results and the minimum requirements for legislation can, with sufficient accuracy, be determined.

The methods and the results produced after a year of research are presented herewith, and the engineering required to produce and maintain these conditions under various requirements of industry in warm summer climates are developed in an engineering survey of the particular problems.

In presenting these findings and the photographs and charts of the year's work, it will be noted that the results cover—and then only partially—factors previously enumerated as items 1 and 3 and reference to part of 9, 10 and 11. This leaves a vast amount of research still to be undertaken, for which both local and government assistance is required. Industry can assist enormously in this work by offering sections of their plants for laboratory test investigation which would illustrate and confirm or augment the work that has already been done. No greater benefit to both capital and labor can possibly result than from a thorough knowledge of the questions which this research has indicated as requiring solution.

REPORT OF TESTS

Of the nine subjects participating in the study, seven were college students of normally good health, ranging in age from 18 to 24 years; the eighth was a

TABLE 1. CHARACTERISTICS AND CONTROL ROOM REACTIONS

SUBJECT		C. J.	D. I.	R. T.	G. D.	B. G.	J. A.	R. S.	L. E.	J. S.
SYMBOL		▲	⊙	●	V	X	■	⊡	·	Δ
AGE - YEARS		30	18	21	23	20	55	23	24	24
WEIGHT - POUNDS		160	150	139	155	138	142	150	150	140
HEIGHT - FT. - IN.		5-6	5-9	5-5	5-10	5-10	5-10	5-6	5-6	5-9
BODY SURFACE AREA SQUARE METERS		1.81	1.81	1.68	1.86	1.78	1.78	1.75	1.75	1.79
BASAL METABOLISM CALORIES/M²HR.		-	40.8	43.0	43.2	34.8	-	49.8	32.3	42.9
BODY TEMPERATURE °F.	MAX.	98.8	99.5	99.1	99.5	99.9	99.7	99.6	99.8	99.4
	AVE.	98.6	98.6	98.5	99.0	99.0	98.9	99.3	99.2	98.7
	MIN.	98.4	98.3	97.8	98.4	98.5	98.0	98.6	98.5	98.4
PULSE RATE BEATS PER MIN.	MAX.	68	98	82	82	100	88	104	93	110
	AVE.	66	72	73	76	77	80	82	86	93
	MIN.	64	63	67	68	68	72	70	81	80
VITAL CAPACITY CU. CM.	MAX.	3296	4161	4326	4841	4532	3090	3811	3955	4700
	AVE.	3193	3729	3935	4388	4367	3090	3667	3955	4408
	MIN.	3090	3399	3543	4202	4079	3090	3502	3955	4120
LEUCOCYTE COUNT PER CU. MM.	MAX.	-	7100	8000	-	8000	-	-	-	8000
	AVE.	-	6567	7767	-	7214	-	-	-	7300
	MIN.	-	6100	7400	-	6300	-	-	-	6400
SYSTOLIC BLOOD PRESSURE MM. HG.	MAX.	117	110	118	124	112	124	110	98	120
	AVE.	110	104	114	117	103	123	107	94	115
	MIN.	108	96	112	102	92	122	104	90	110
DIASTOLIC BLOOD PRESSURE MM. HG.	MAX.	80	75	78	81	78	80	74	66	79
	AVE.	77	69	77	73	71	77	70	65	77
	MIN.	75	66	76	64	62	75	66	62	72

55-year-old carpenter; while the ninth was a 30-year-old colored man usually employed in the electrical manufacturing industry. Before accepting a prospective subject for the study he was subjected to a thorough physical examination by a practicing physician.

In Table 1 is included the following general information: the initials of the subjects, the symbols used in analyzing and plotting their data, their pertinent physical characteristics as found in the preliminary examination, and some of their physiological reactions when seated at rest in a fairly com-

fortable, well-ventilated but not cooled or air conditioned control room. The clothing worn by the subjects included light-weight undershirts without sleeves, shorts, light trousers, socks, and shoes.

TEST PROCEDURE

The subjects reported for test about 8:30 a.m. and entered the control room, where they were seated at rest for 45 min or longer. It was assumed that their physiological reactions would readjust themselves during this time. The readjustment would be from equilibrium with the morning activity including coming to work, to an equilibrium with a normal resulting from rest in a comfortable atmosphere. During occupancy in the control room, observations for each subject were made of body temperature, pulse rate, vital capacity, leucocyte count, blood pressure, weight, body and forehead perspiration, and feeling of warmth. Some of the observations were made on a number of occasions during the 45 min period, while other observations such as leucocyte counts, which were difficult to make and which required special technique and assistance, were made less frequently.

At the close of the 45 min period in the control room, the subjects entered the test chamber where they engaged in a predetermined physical activity for four hours, excepting for very hot test conditions when it was necessary to shorten this period. During their occupancy in the test room frequent observations of the physiological reactions of the subjects were made and recorded, in order to determine the variations of these reactions from those established as normal in the control room. A final observation of all the pertinent physiological reactions was made just before each subject left the test room. After leaving the test room when particularly hot conditions had been maintained and the body temperature and pulse rate had been considerably elevated, the subjects were required to re-enter the relatively comfortable atmosphere in the control room until their reactions became substantially normal. Then they usually had a shower and returned to their everyday activity.

Tests were conducted five days a week with little interruption from July 7, 1938 to September 6, 1938. Provisions were made for having four subjects in the test room, and with few exceptions this number participated in each test. Having nine subjects available allowed some modification of the frequency of tests on any one individual. The seven students participated in the greater number of tests, and only a few tests were made with the two older men.

PHYSIOLOGICAL OBSERVATIONS

The plans for the study included observation of all the physiological reactions of the subject which might respond in any measurable degree both to his activity and to the atmospheric conditions maintained in the control room. All of the physiological reactions listed in Table 1, as well as the loss in weight, the degree of sensible perspiration on the forehead and body, and the feeling of warmth, were observed for all the subjects in all of the tests. The metabolic rate, the blood pressure, and the leucocyte count, all of which re-

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quired special equipment and technique, were not always taken for all the subjects in the various tests.

A practicing physician with special training and experience in physiological research, who was connected with the Department of Industrial Hygiene of the School of Medicine of the University of Pittsburgh, was at all times available for consultation and aid in carrying on the work. He actually participated in the more difficult physiological measurements in many cases, but when he was not available a senior medical student of the University working under his supervision aided in making these observations.

The activity during the morning, including breakfast and coming to work, and in some cases the somewhat elevated temperature in the control room may account for some of the slightly elevated body temperatures, pulse rates and leucocyte counts. The body temperatures were observed rectally with clinical mercurial thermometers, and it was found that the averages were much nearer 99 F than the conventionally assumed 98.6 F. In this connection it should be kept in mind that body temperatures observed rectally are usually somewhat higher than those observed orally. The values given in Table 1 also serve to emphasize the fact that there is considerable variation in the body temperature of different normal persons, and also in the body temperature of the same person at different times. As seen from Table 1, the group gave a minimum to maximum range of from 97.8 to 99.9 F.

The pulse rate was observed by counting the wrist pulse with a suitable timepiece. The vital capacity, expressed in cubic centimeters, was determined as the maximum volume of expired air which the subject could blow into a calibrated spirometer at one exhalation at the time of observation. It will be observed that this varied considerably in the control room for the different subjects and to some extent for the same subject on different days.

The doctor or medical student always made the leucocyte count, and it was found that the normal values taken in the control room and included in Table 1 were a little high for what is generally accepted as normal. However, this may be due to the fact that these observations were usually taken within 1½ hour after breakfast, followed by the exertion incident upon coming to work. The systolic and diastolic blood pressures were observed in the manner usually used by doctors, with instruments including a mercury manometer.

The basal metabolic rate and the metabolism both while seated at rest and while working were determined by the indirect method used in earlier laboratory studies of heat and moisture loss from the body. Between the wearer's mask and the apparatus was provided a 4-ft hose connection, which allowed the subject sufficient freedom for the work required of him. This work was continued uninterrupted with his wearing the mask for several minutes before and during the actual determination.

The loss in weight was determined by weighing the subject on scales accurate to ½ oz, first in the control room just before his entering the test room and again before his leaving the test. Since a record was kept of water consumed and urine excreted during this period, the loss in weight was corrected accordingly. Hence, the loss in weights given are net loss due to perspiration and carbon expired as carbon dioxide.

The observation and recording of condition of perspiration on forehead and body were made in accordance with a scale of degree of perspiration adopted

and used in earlier studies at the Laboratory. According to this scale, entire freedom from sensible perspiration is given the index of zero; clamminess, "1"; dampness, "2"; wetness, "3"; and running down or soaking through clothing of perspiration, "4."

The degree of feeling of warmth or comfort was observed also in accordance with a scale used in earlier studies at the Laboratory. According to this scale, ideal comfort was expressed as "4", slightly warm and slightly cool conditions by "5" and "3", respectively, decidedly warm and decidedly cold conditions by "6" and "2", respectively. Since all of the tests in this study were at temperatures at or above those giving a feeling of ideal comfort, only indices of "4", "5" and "6" were necessarily used. The measure of the degree of comfort or perspiration by these arbitrary indices is rather difficult and may be subject to considerable variation or possible error in observations, due to the observer's misconception of just what degree of reaction a given index number is to represent, and to his inability to interpret this degree properly at times. Particularly in the case of the comfort reactions, difficulty was experienced in this study; for frequently, the subjects were working in hot atmospheres as compared to earlier studies in which the subjects had been at rest in relatively comfortable atmospheres, or atmospheres only slightly too warm or too cool for comfort. Accordingly, the index previously used was slightly modified in this study in its interpretation. A comfort index of "4" in this study, as in the earlier ones, represented ideal comfort. It was rather surprising, however, to find that the subjects in this study while working in atmospheres of 77½ deg ET, and sometimes as high as 82 deg, said they were comfortable; whereas in earlier studies the subjects were rarely found to be comfortable during the summer time in atmospheres of 75 deg ET, and more often the upper limit of comfort in the earlier studies was found to be around 73 or 74 deg ET. This difference may be accounted for in one of two ways. It is possible, and even probable, that the extra body surface area exposed to normal heat loss to the atmosphere for a person standing is more appreciable than for a person seated; further, the added effective air motion over the surface of the body for a person standing up and moving about may actually serve to increase the rate of heat loss to an extent greater than the added rate of heat production in the body by metabolism due to the activity of movement. The second reason may be psychological in that a person engaged in some activity is less likely to have his attention directed to a feeling of slight discomfort caused by higher atmospheric temperatures, as in the case of a person at rest.

A comfort rating of "5" in this study was made to include observations of rather decidedly warm but still bearable conditions, while "6" was made to include conditions which were very hot and more or less unbearable.

TEST CONDITIONS

The study included observations of the physiological reactions of the subjects while at light work in atmospheres ranging from 77½ to 92½ deg ET, and for 60, 75 and 90 per cent relative humidity. At each of the three humidities studied, tests were made at approximately 77½, 80, 82½, 85, 87½, 90 and 92½ deg ET. For at least an hour before the subjects entered the test room

the air temperature and moisture content were kept constant at the predetermined values within the accuracy of one degree dry-bulb temperature and one per cent relative humidity. The wet- and dry-bulb temperatures, recorded automatically, were also observed by a fan-operated psychrometer consisting of wet- and dry-bulb thermometers so arranged that the fan drew the air over them at a velocity of approximately 2000 fpm. Both the recording and the previously mentioned indicating instrument were frequently checked by a sling

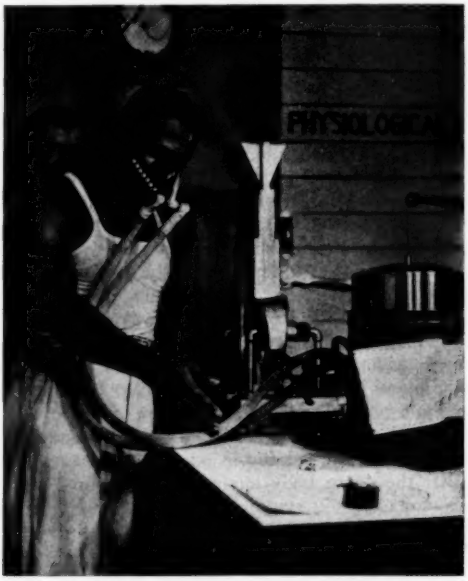

FIG. 3. VIEW OF TEST TABLE WITH CHANCE MACHINE SHOWING METABOLISM DETERMINATION ON SUBJECT, C. J.

psychrometer. The air motion within the test room always ranged from 30 to 40 fpm, which was the movement necessary to maintain uniform conditions. In this study no attempt was made to include the effect of air movement on the physiological reactions of the subjects.

ACTIVITY OF THE SUBJECTS

This study was entirely limited to the physiological reactions of workers engaged in what was called *light work*. The type of work agreed upon by the Committee required the subject to be on his feet approximately three-quarters of the time, to move about performing light tasks entailing some

but not severe muscular activity; and above all the work required that the subject be alert and attentive to his task. This attentiveness of the subject was the most difficult to provide for, since it is a well-recognized fact that a person performing a routine physical effort may soon learn to perform that task when he is practically asleep mentally, or when his mind may be occupied with thoughts entirely extraneous to the work performed. To make attention an essential part of the task required that some element of chance should determine the particular duties that the subject would be called upon to do and the time when they must be done.

A chance machine shown in Fig. 3 was developed into which each of the subjects dropped a small ball, or marble, each subject's ball being distinguished

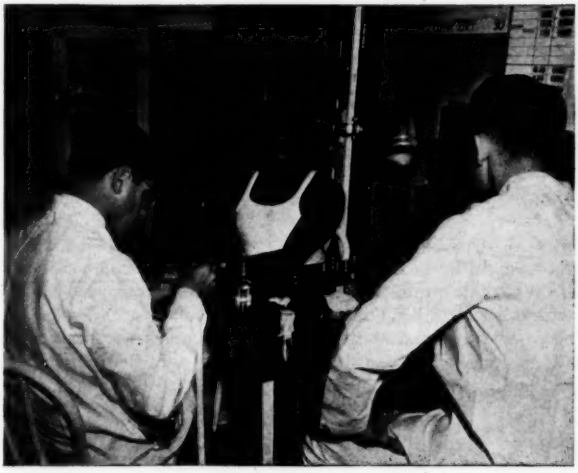

FIG. 4. VIEW OF SUBJECT HAVING BLOOD SAMPLE TAKEN AND ANALYZED FOR DETERMINATION OF LEUCOCYTE COUNT

by color. After this ball was dropped into the machine, the chance operation of the machine delivered it within the interval of ½ and 2½ min. Therefore, during this time the subject was required to be attentively watching the exit of the machine to see when his particular ball was emitted. When that subject's ball was emitted he immediately threw two dice, one of which determined the particular weight of a series, ranging from ¼ to 1¾ lb, that he had to pick up from the table, while the second dice determined the new position on the table where he had to place the weight. Since there were six places on the table and each subject had six differently numbered weights, considerable attention was required in choosing the right weight and putting it in the right place. When replacing the weights each subject was required to move around the table in performing his task rather than reaching across the table to do so. When these tasks were performed the subject was required to record the time,

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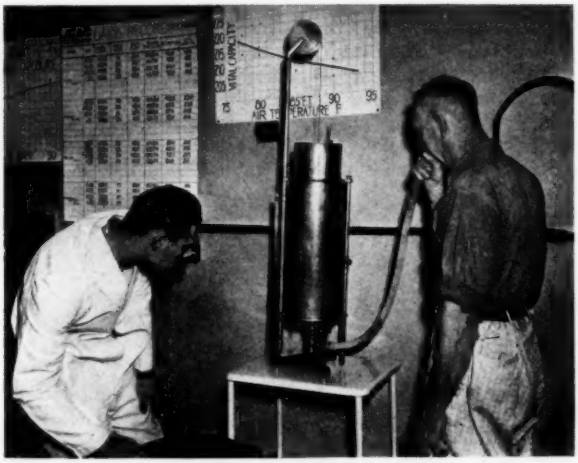

FIG. 5. DETERMINING VITAL CAPACITY OF SUBJECT, J. A.

the particular weight he picked up, and the location where he placed it. While performing these duties, the subject also observed and recorded various physiological reactions such as his body temperature, pulse rate, degree of perspiration, feeling of warmth, and at intervals blood samples were taken for analysis as illustrated in Fig. 4 and also the vital capacity was determined as shown in Fig. 5. Each subject was weighed during the test to indicate perspiration loss as illustrated in Fig. 6.

TEST DATA AND RESULTS

In Fig. 7 are plotted some of the physiological reactions for one of the subjects (D. I.) during a 4-hour test in an air condition of 90 deg ET at 75 per cent relative humidity. This subject was at rest in the control room from 9 to 10 a.m., at which time he entered the test room and worked for 4 hours, or until 2 p.m. Body temperatures and pulse rates were observed every 15 min in the control and a little less often in the test room. The other physiological observations, because of the difficulty incident to making them, were taken at less frequent intervals.

In addition to this subject's (D. I.) physiological reactions as plotted in Fig. 7, his metabolic rate before leaving the control room was 40.8 calories per square meter per hour, and his average systolic and diastolic blood pressures were 104 and 69, respectively.

It will be observed from Fig. 7 that both the body temperature and the pulse rate show a progressive change with time of exposure in the hot atmosphere. This rise in body temperature is logically expected to occur

since body temperature must show a proportional rise with accumulated heat. The accumulation of heat is in turn a function of the rate of heat produced in the body and the rate of heat loss from the body to the atmosphere and surroundings. About the only factor which is likely to change this porportional relationship is the slight increase of metabolism with rise in body temperature. This relationship has been credited to the purely chemical law of Van't Hoff, which states that most chemical reactions are a function of and increase with temperature. This is contrary to the best interests of the body which would dictate that with rise in body temperature the metabolic rate, or rate of heat production, should decrease. In other words, the point where metabolism increases with rise in body temperature or with severity of a hot atmosphere represents the initial breakdown of the processes of life, or the adjustment of the internal reactions of the body to meet the external environment. Then it becomes a question of a further small elevation in the severity of the hot condition or length of time of exposure before death occurs. It is interesting to point out that the progression of physiological changes is an accelerative process after the maximum temperature at which the body can adjust its internal reactions to its environment has been reached.

It will be noticed that change in body temperature and also the pulse rate do not follow straight line relationships with time. The body temperature changes rapidly immediately after entering the test room and tends to become a constant with increased length of exposure. This is because the rise in body temperature in the constant atmospheric condition decreases the temperature difference between the body and its environmental air. If a person were able to exist satisfactorily at a sufficiently elevated body temperature, his physiological mechanisms would consequently establish a higher temperature

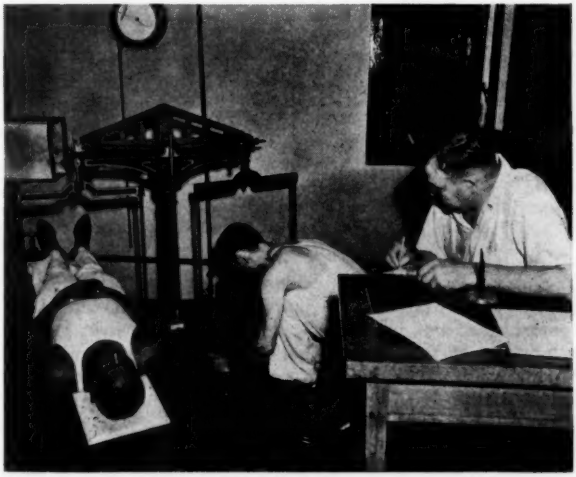

FIG. 6. WEIGHING SUBJECT, C. J., FOR PERSPIRATION LOSS

level at which equilibrium could be maintained. This adjustment, however, can only be made within rather small limits compatible with life; and even at lesser elevations in temperature where death or collapse would not occur, the individual ceases to function normally.

Increase in pulse rate is not necessarily a quantitative change resulting directly from exposure, but represents a physiological reaction resulting from nerve stimulus. If psychic factors are eliminated, the stimulus causing an

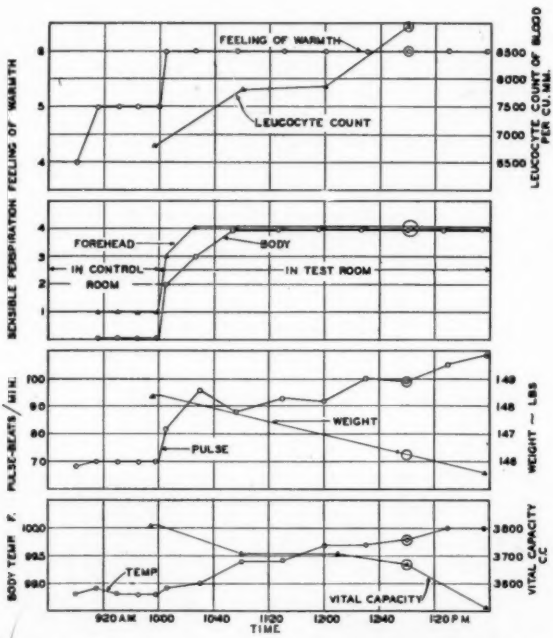

FIG. 7. PHYSIOLOGICAL RESPONSE OF D. I. AT REST IN CONTROL ROOM AND WHILE WORKING IN AN AIR CONDITION OF 90 DEG ET AND 75 PER CENT RELATIVE HUMIDITY

increase in pulse rate is apparently more or less proportional to the rise in body temperature, for increase in pulse rate parallels quite closely rise in body temperature, except in unusual diseased states. It has been shown in other work, however, particularly in the application of fever therapy,[2] that if the patient can be kept quiet or unexcited, either through his own self control or due to the application of a sedative, a rise in body temperature may not be accompanied by the same increase in pulse rate, except where there is already a high basal metabolic rate or where there may be some physical debility.

Plots or tabulations of the several physiological reactions similar to those

in Fig. 7 were made for each individual subject in all of the tests made at 60, 75 and 90 per cent relative humidity. The several reactions after 3 hours of exposure are plotted in Figs. 8 to 21, and the points in each case were taken from the plotted curves similar to those in Fig. 7, or from calculation based upon the rate of change where fewer observations were taken, as in the

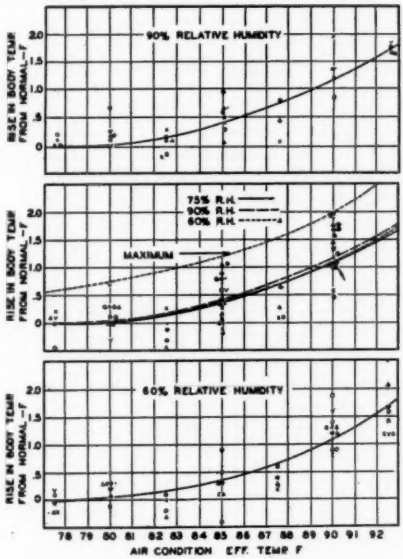

FIG. 8. RELATION BETWEEN RISE IN BODY TEMPERATURE FROM NORMAL DURING 3-HOUR EXPOSURE AND THE EFFECTIVE TEMPERATURE OF THE AIR CONDITION IN THE TEST ROOM FOR 60, 75 AND 90 PER CENT RELATIVE HUMIDITIES. TEST POINTS FOR INDIVIDUAL SUBJECTS[a] APPLY TO HEAVY SOLID CURVES

case of vital capacity and blood pressure. As an example, if the vital capacity was found to be 4000 cc before entering the test room and 3650 cc after 3½ hours of exposure, the hourly rate of change was assumed to be 100 cc, and change in the vital capacity in 3 hours was assumed to be 300.

The relation between the rise in body temperature and the effective temperature, dry-bulb temperature and wet-bulb temperature of the atmosphere for the three humidities studied are plotted in Figs. 8, 9 and 10, respectively. The results of tests at the three different relative humidities are plotted sepa-

[a] See Table 1.

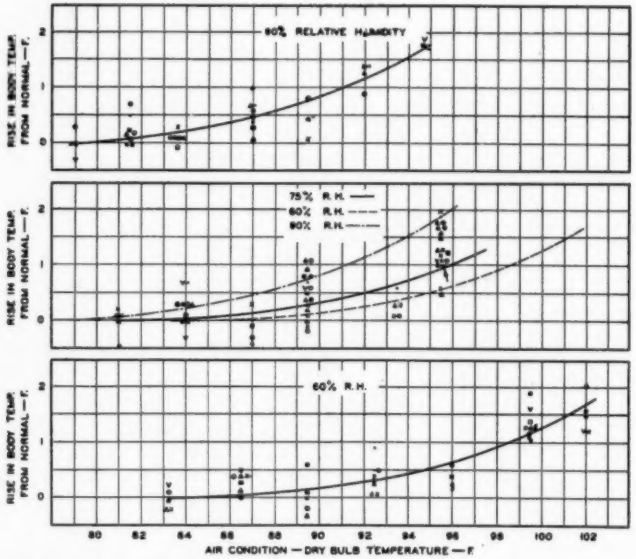

FIG. 9. RELATION BETWEEN RISE IN BODY TEMPERATURE FROM NORMAL DURING 3-HOUR EXPOSURE AND THE DRY-BULB TEMPERATURE OF THE AIR CONDITION IN THE TEST ROOM FOR 60, 75 AND 90 PER CENT RELATIVE HUMIDITIES. TEST POINTS FOR INDIVIDUAL SUBJECTS [b] APPLY TO HEAVY SOLID CURVES

[b] See Table 1.

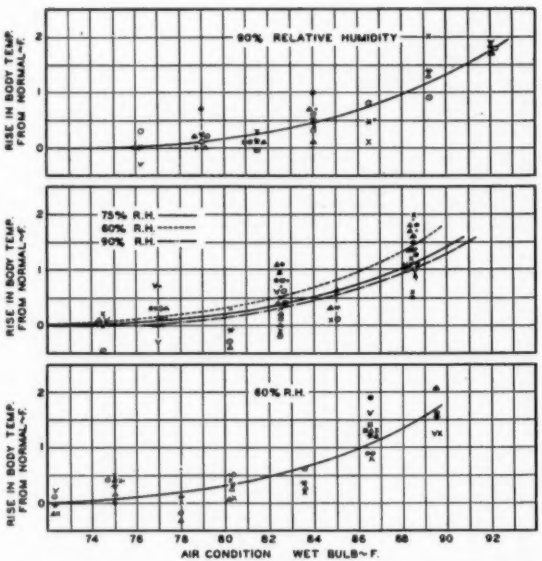

FIG. 10. RELATION BETWEEN RISE IN BODY TEMPERATURE FROM NORMAL DURING 3-HOUR EXPOSURE AND THE WET-BULB TEMPERATURE OF THE AIR CONDITION IN THE TEST ROOM FOR 60, 75 AND 90 PER CENT RELATIVE HUMIDITIES. TEST POINTS FOR INDIVIDUAL SUBJECTS [b] APPLY TO HEAVY SOLID CURVES

rately. In each case the solid line curves are drawn for the plotted test points for the different subjects (see Table 1) after 3 hours exposure. The resulting curves for 60 per cent and 90 per cent relative humidity tests are superimposed on the 75 per cent relative humidity section of the chart. It will be noted that when temperature rise is plotted against effective temperature there is little separation of the curves for the different relative humidities. In the plot

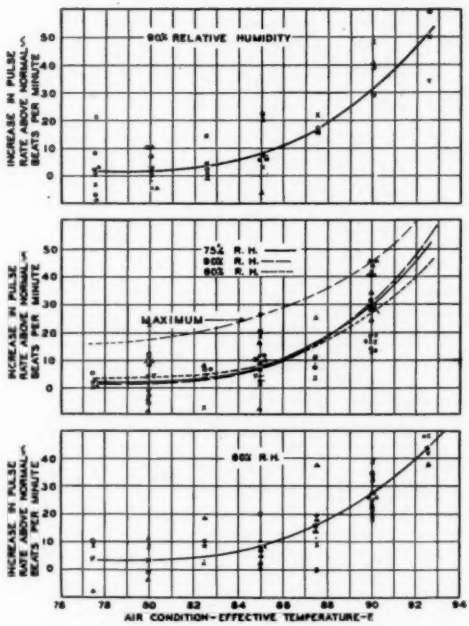

FIG. 11. RELATION BETWEEN INCREASE IN PULSE RATE FROM NORMAL DURING 3-HOUR EXPOSURE AND THE EFFECTIVE TEMPERATURE OF THE AIR CONDITION IN TEST ROOM FOR 60, 75 AND 90 PER CENT RELATIVE HUMIDITIES. TEST POINTS FOR INDIVIDUAL SUBJECTS* APPLY TO HEAVY SOLID CURVES

against wet-bulb temperature there is some separation while for the dry-bulb plot there is a very significant separation of the three curves. It is of interest to note that when plotted against wet-bulb temperature, the separation is such as to give a series from 60 to 90 per cent relative humidity from the upper curve to the lower, while when plotted against dry-bulb the curves are in the reverse order. This is as should be expected from the relation of effective temperature to wet-bulb and dry-bulb temperatures; that is, for any atmospheric

* See Table 1.

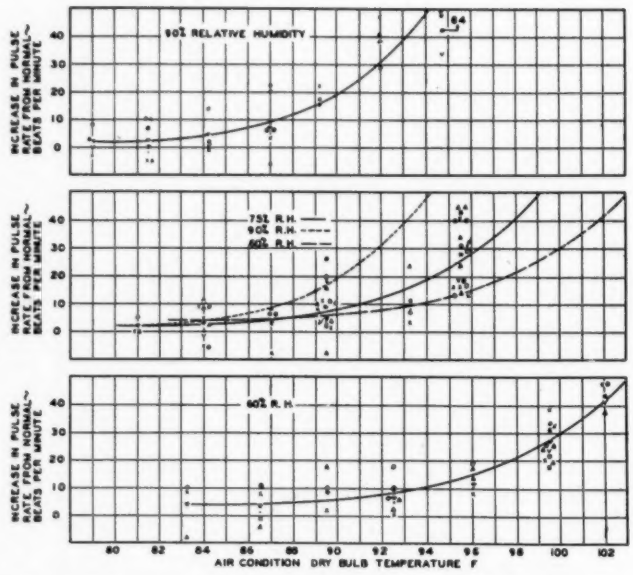

FIG. 12. RELATION BETWEEN INCREASE IN PULSE RATE FROM NORMAL AND THE DRY-BULB TEMPERATURE OF THE AIR CONDITION IN THE TEST ROOM FOR 60, 75 AND 90 PER CENT RELATIVE HUMIDITIES. TEST POINTS FOR INDIVIDUAL SUBJECTS[4] APPLY TO HEAVY SOLID CURVES

[4] See Table 1.

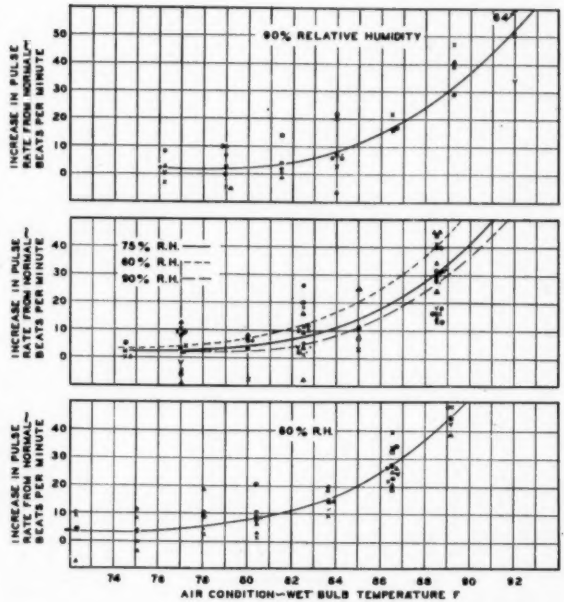

FIG. 13. RELATION BETWEEN INCREASE IN PULSE RATE FROM NORMAL AND THE WET-BULB TEMPERATURE OF THE AIR CONDITION IN THE TEST ROOM FOR 60, 75 AND 90 PER CENT RELATIVE HUMIDITIES. TEST SUBJECTS[4] APPLY TO HEAVY SOLID CURVES

condition giving a certain effective temperature at any relative humidity below 100 per cent the wet-bulb temperature is lower and the dry-bulb temperature is higher than the effective temperature.

The considerable variation for the rise in body temperature of different subjects and for the same subject in different tests in the same air condition indicates that variation in the physiological reactions of individuals or the same individual may be expected at different times. This variation under different test conditions is not much different from the variation in body temperature after ¾ hour of rest in the control room, as given in Table 1.

The increase in pulse rate for all subjects after 3 hours exposure in all tests at 60, 75 and 90 per cent relative humidity is plotted against effective

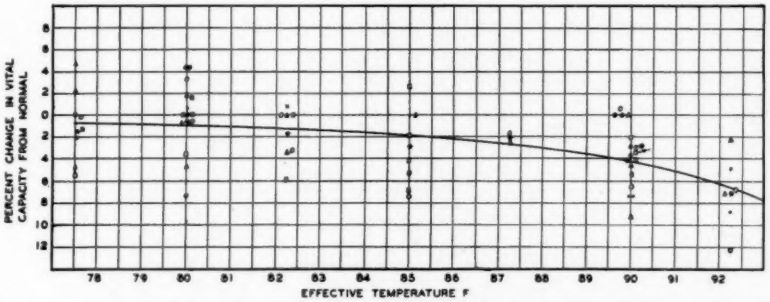

FIG. 14. PER CENT CHANGE IN VITAL CAPACITY AFTER 3 HOURS OF WORK IN ATMOSPHERIC CONDITIONS OF 60, 75 AND 90 PER CENT RELATIVE HUMIDITIES AND THE EFFECTIVE TEMPERATURES INDICATED. CURVE SHOWS CHANGE FOR THE THREE HUMIDITIES

temperature, dry-bulb and wet-bulb temperatures, respectively, in Figs. 11, 12 and 13. In each case the test points for the different subjects fit the solid line curve representing a 3-hour exposure. The curves for 60 and 90 per cent relative humidity are in each case superimposed on that for 75 per cent, in order to make possible a comparison of the effects of tests at the different humidities.

The relation between percentage change in the vital capacity of the subjects and effective temperature of the atmospheric condition for all subjects in all tests in which this observation was made is given in Fig. 14. No discernible variation in the reaction for tests at different relative humidities was indicated, and consequently the points for all three humidities are plotted in this single chart, which shows the changes taking place during a 3-hour period. While there is a rather definite decrease in the vital capacity after 3 hours exposure for the hotter atmospheres, the change is not very significant in that the variation between individuals for the same condition on the same day and for the same individual on different days is greater than the average change due to the severity of the atmospheric condition.

The leucocyte count of the blood in white cells per cubic millimeter for all

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subjects in all tests in which this observation was made is plotted in Fig. 15. Since this observation was rather difficult to make, in that it required special equipment and skill of manipulation, only a relatively few observations were made. In these tests the count was made in the control room just before entering the test room, and again after 3 hours exposure at work in the hot condition. The observations in the control room were all made on days when the effective temperature maintained was 75 deg. The curve shows a definite increase in the count after 3 hours exposure in the hot conditions. The leucocyte count in the control room is a little higher than is generally given for normal people in normal atmospheres, which fact may be due to its being taken within a couple of hours after the subject's rising in the morning, eating

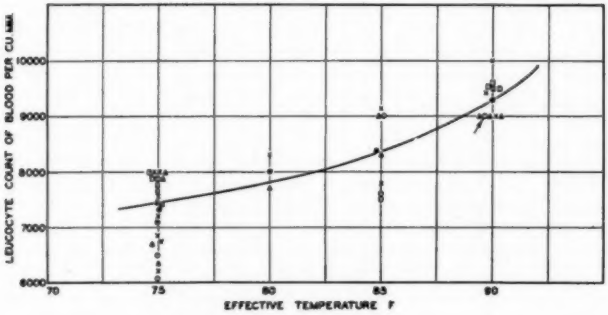

FIG. 15. CHANGE IN LEUCOCYTE COUNT OF BLOOD AFTER 3 HOURS OF WORK IN ATMOSPHERIC CONDITIONS OF 60, 75 AND 90 PER CENT RELATIVE HUMIDITIES AND THE EFFECTIVE TEMPERATURES INDICATED. CURVE SHOWS CHANGES FOR THE THREE HUMIDITIES

breakfast and coming to work; the latter task frequently involved a walk of some distance. An average of 6000 resulting from variations of from 5000 to 7000 would be more expected from normal, inactive persons in a comfortable atmosphere. The increase in the count after 3 hours of working in the hotter atmosphere is definite and probably of considerable importance. However, unlike elevation in body temperature and increase in pulse rate, no definitely harmful significance can be placed on the increased leucocyte count. But it should be kept in mind that an increase in the count generally accompanies any kind of infection in the body or an elevation in the body temperature. This increase is generally considered as a preparation of the body to cope with infection.

The blood pressure as observed for all subjects in all tests for the three humidities is plotted in Fig. 16. While there is indicated a consistent rise in systolic and a less pronounced fall in the diastolic pressure for exposures in atmospheres above 87 deg ET, these changes in the average curves are not very significant when compared with the wide variation in the values for

different individuals on the same day or for the same individual on different days. These curves are, however, consistent with facts observed in the application of fever therapy,[2] where experience shows an increase in the systolic and a decrease in the diastolic pressures with elevation in body temperature due to hot atmospheres. A certain maximum increase in the systolic can occur, after which a further rise in the body temperature may result in a decrease in systolic pressure, which if it continues to fall may result in collapse. Upon complete collapse the systolic pressure may even fall until it approaches the diastolic pressure. For this latter condition of collapse the pulse pressure may be extremely low.

The variations in metabolism with activity and exposure for the different subjects are shown in Fig. 17. The rates of metabolism for basal, seated at

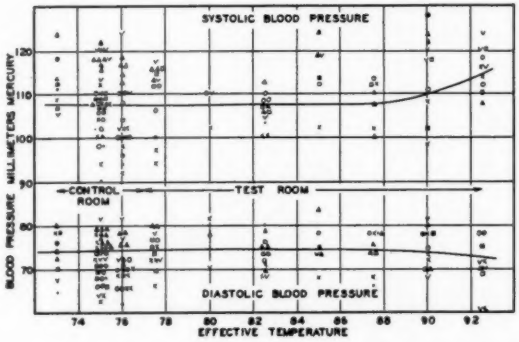

FIG. 16. CHANGE IN SYSTOLIC AND DIASTOLIC BLOOD PRESSURE AFTER 3 HOURS OF WORK IN ATMOSPHERIC CONDITIONS OF 60, 75 AND 90 PER CENT RELATIVE HUMIDITIES AND THE EFFECTIVE TEMPERATURES INDICATED. CURVES SHOW CHANGES FOR THE THREE HUMIDITIES

rest, and light work are for the subjects in the tests in this study. On mornings when the subject had not eaten breakfast, basal observations were made on him after he had lain at complete rest for ¾ hour immediately following arrival at the Laboratory. On other mornings the *seated at rest* observations were made on subjects toward the end of the ¾ hour period in the control room, while the observations for *standing at light work* were made in the test chamber after 3 hours exposure. The points plotted are for tests at the three humidities, there being no significant variation either for tests at different humidities, effective temperatures or lengths of time of exposure. This is in accordance with observations made in earlier work at the Laboratory,[3] in which usually no change in metabolism was found upon exposure to hot atmospheres, excepting that accompanying a considerable elevation in the body temperature. The points giving the metabolic rate at moderate work and heavy work were determined in earlier Laboratory studies.[4] These rates of

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work consisted of raising and lowering, by means of rope and pulley, a 40-lb weight 7½ and 11 times per minute, respectively, through a distance of 5 ft. Moreover, these rates of work were performed continuously for 3 or 4 hours. The higher rate was designated in the earlier study as *heavy work*, which term is repeated here. Table 2 shows rates of work having as high as 4800 Btu per hour for the resulting metabolism for an average sized man, but it is doubtful if such high rates could be continued for an extended length of time.

The rate of loss in weight of the different subjects while engaged in work under the different atmospheric conditions is plotted against effective temperature in Fig. 18. Points for all relative humidities are included and these

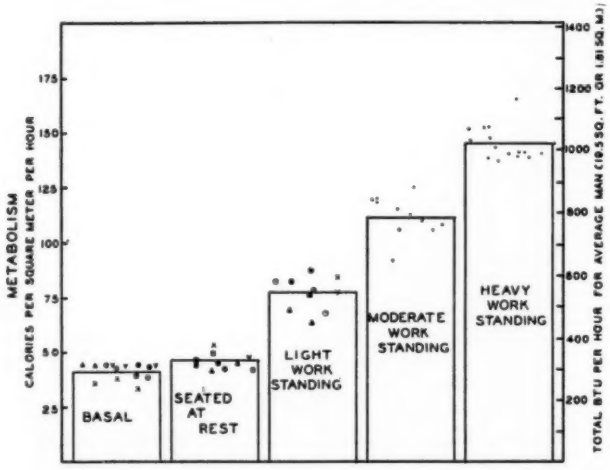

FIG. 17. RELATION BETWEEN METABOLIC RATE AND DEGREE OF ACTIVITY

determine the solid line curve. The broken line curves for *at rest* and *heavy work* are from earlier Laboratory studies [1, 5] and will be discussed later in this paper.

Loss in weight of a person in a hot atmosphere, or in fact to a lesser extent in any atmosphere, is due largely to the evaporation of moisture. The major part of this evaporation takes place from the surface of the body, while a lesser portion takes place from the respiratory tract. To a still lesser extent, a continued loss in weight occurs as the result of the exhalation of carbon in the form of carbon dioxide, and this loss is approximately proportional to the metabolic rate. Earlier Laboratory studies [3, 4] have shown that this carbon loss may range from 0.3 oz per hour for a person at rest to 0.8 oz per hour for a person at heavy work. It will be seen, therefore, that it is a rather insignificant part of the weight loss shown.

An analysis of the data indicates a very slight, though not significant, change

TABLE 2—RELATION BETWEEN METABOLIC RATE AND ACTIVITY[a]

ACTIVITY	HOURLY METABOLIC RATE FOR AVERAGE PERSON OR TOTAL HEAT DISSIPATED BTU PER HOUR	HOURLY SENSIBLE HEAT DISSIPATED BTU PER HOUR	HOURLY LATENT HEAT DISSIPATED BTU PER HOUR	MOISTURE DISSIPATED PER HOUR	
				Grains	Pounds
Basal [11]	291	145	145	979	0.140
Seated at Rest [11]	384	225	159	1,070	0.153
Reading Aloud (Seated)[17]	420	225	195	1,316	0.188
Standing at Rest [11]	431	225	206	1,390	0.199
Hand Sewing (Seated)[17]	441	225	216	1,458	0.208
Knitting 23 Stitches per Min. on Sweater[17]	462	225	237	1,600	0.229
Dressing & Undressing[17]	468	225	243	1,640	0.234
Tailor[13]	482	225	257	1,740	0.248
Singing[17]	486	225	261	1,762	0.252
Office Worker Moderately Active[12]	490	225	265	1,790	0.256
Light Work Standing[11, b]	549	225	324	2,187	0.312
Typewriting Rapidly[17]	558	225	333	2,248	0.321
Ironing with 5 lb Iron[17]	570	225	345	2,329	0.333
Dishwashing—Plates, Bowls, Cups, Saucers[17]	600	225	375	2,530	0.362
Clerk Moderately Active Standing at Counter[12]	600	225	375	2,530	0.362
Book Binder[13]	626	225	401	2,710	0.387
Shoemaker[13]	661	225	436	2,940	0.420
Sweeping Bare Floor 38 Strokes per Min[17]	672	229	443	2,990	0.427
Pool Player[12]	680	230	450	3,040	0.434
Walking 2 mph [14, 15], Light Dancing[14, 15]	761	250	511	3,450	0.493
Light Metal Worker (at Bench)[13]	862	277	585	3,950	0.564
Painter of Furniture (at Bench)[13]	876	280	596	4,020	0.575
Carpenter[17]	954	307	647	4,367	0.624
Restaurant Serving[12]	1,000	325	675	4,556	0.651
Pulling Weight[11, c]	1,041	335	708	4,745	0.677
Walking 3 mph[14]	1,050	339	711	4,750	0.679
Walking 4 mph, Active Dancing, Roller Skating[14, 15]	1,390	452	938	6,330	0.904
Walking Down Stairs[17]	1,444	467	977	6,595	0.942
Stone Mason[13]	1,490	490	1,000	6,750	0.964
Bowling[12]	1,500	490	1,010	6,820	0.974
Man Sawing Wood[13]	1,800	590	1,210	8,170	1.167
Swimming[17]	1,986	...	...	...	...
Running 5.3 mph[17]	2,268	...	...	...	...
Walking 5 mph[14]	2,330	...	...	...	...
Walking Very Fast 5.3 mph[17]	2,580	...	...	...	...
Walking Up Stairs[17]	4,365	...	...	...	...
Maximum Exertion, Different People	3000-4800	...	...	...	...

[a] Reference numbers refer to bibliography.
[b] This study.
[c] Pulling 40-lb weight 5 ft over pulley 11 times per minute.

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in the weight loss for different relative humidities at the same effective temperature. As an example, the curves so plotted at 90 deg ET, where the maximum variation occurs, indicate losses of 7, 8½ and 10½ oz per hour at 90, 75 and 60 per cent relative humidities, respectively. While this difference in weight loss with relative humidity is not significant when compared with the wide variation for different individuals on the same day or the same individual on different days, it is nevertheless of interest in considering thermal exchanges between the body and the atmospheric environment.

The relations between perspiration on the body and forehead of the subjects after 3 hours exposure, and the effective temperature of the test room are plotted in Figs. 19 and 20. While there is considerable scattering of points at any given effective temperature for different subjects on the same day and, to a lesser extent, for the same subject on different days, the mass of data

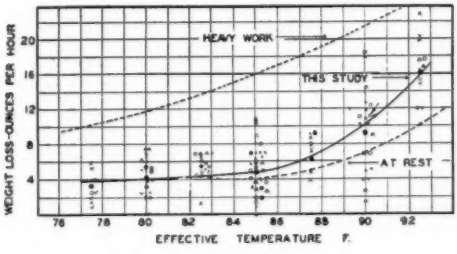

FIG. 18. RELATION BETWEEN WEIGHT LOSS IN OUNCES PER HOUR AND THE EFFECTIVE TEMPERATURE OF THE AIR CONDITION FOR THE SUBJECTS IN THIS STUDY. CURVES FOR PERSONS AT REST AND AT HEAVY WORK ALSO SHOWN

shows a very positive increase in the intensity of sensible perspiration with severity of the hot condition. The scattering of the points may in some measure be accounted for by individual variation, and also to the inability of the individual subject to properly evaluate the condition of his perspiration in terms of the index used. This difficulty was greatest when trying to interpret the difference between *clammy, damp,* and *wet* in the case of forehead perspiration. At the same effective temperature there was no distinguishable variation in the degree of perspiration on either the body or forehead with the different relative humidities. Also, there was relatively little variation in the degree of perspiration with time of exposure after the first one-half to one hour of exposure.

The feeling of warmth of the subjects for different atmospheric conditions is plotted in Fig. 21. It is difficult to explain the difference in indicated effective temperature for comfort conditions in this study and previous ones.[6, 7, 8] Whereas a subject in this study felt comfortable at an effective temperature of 82 deg, a person seated *at rest* in former studies required an effective temperature of about 72 deg. Both psychological and purely physical effects come

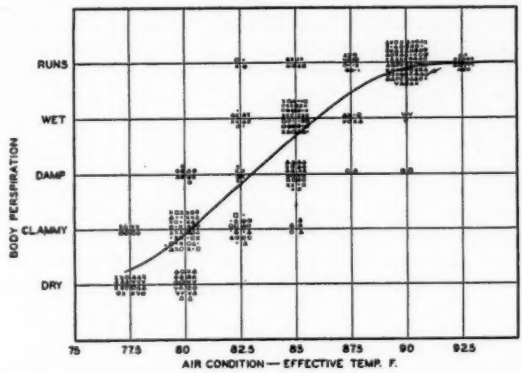

FIG. 19. RELATION BETWEEN DEGREE OF PERSPIRATION ON BODY AFTER 3 HOURS EXPOSURE AND THE EFFECTIVE TEMPERATURE OF THE AIR IN THE TEST ROOM; FOR ALL SUBJECTS IN TESTS AT 60, 75 AND 90 PER CENT RELATIVE HUMIDITIES

into play in interpreting feeling of comfort, and it is problematical as to the extent to which either may account for the difference. As mentioned previously, the physical factors of larger exposed body surface area and resultant increase in air movement must have some additional effect in increasing the rate of heat transfer. Whether these purely physical changes account for the

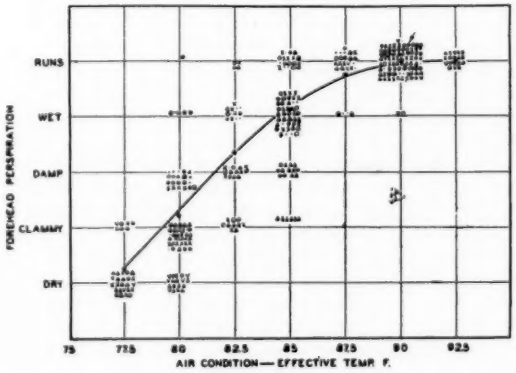

FIG. 20. RELATION BETWEEN DEGREE OF PERSPIRATION ON FOREHEAD AFTER 3 HOURS EXPOSURE AND THE EFFECTIVE TEMPERATURE OF THE AIR IN THE TEST ROOM; FOR ALL SUBJECTS IN TESTS AT 60, 75 AND 90 PER CENT RELATIVE HUMIDITIES

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entire difference as between comfort at 82 deg ET for persons in these tests, and about 72 deg ET for persons seated at rest in previous tests, or whether the psychological interpretation also has its effect is difficult to determine. It is believed, however, that the purely physical factors predominate. In earlier Laboratory studies [4] it was rather definitely proven in a qualitative, although not in a quantitative manner, that in the same atmosphere sensible heat loss from the body increased for persons at work over that for persons at rest.

The solid-line curves in Fig. 22 show the relation between the effective temperature of the atmospheric condition, for lengths of exposure of ½ hour, 1 hour, 2 hours, 3 hours, and 4 hours, and both the rise in body temperature and the increase in pulse rate. The 3-hour curves are the averages of the curves for the three humidities in Figs. 8 and 11. It will be seen that they

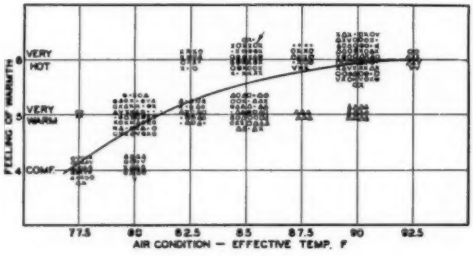

FIG. 21. RELATION BETWEEN FEELING OF WARMTH AND THE EFFECTIVE TEMPERATURE OF THE AIR IN THE TEST ROOM FOR ALL SUBJECTS IN TESTS AT 60, 75 AND 90 PER CENT RELATIVE HUMIDITIES

form a rather definite series, exemplifying the fact mentioned in connection with Fig. 7, that the rate of rise in body temperature during exposure in a given hot condition decreases with the length of exposure. As an example, the rise during the first ½ hour is greater than the rise during the fourth hour. It should be emphasized in this connection that the indicated rates of rise for 3- and 4-hour exposures in the more severe conditions were based upon the rate of rise during a shorter period; that is, the subjects were not kept in a 92 deg ET atmosphere for 4 hours, but for this condition the time was always limited to 3 hours and frequently to a shorter period.

In discussing the data thus far, consideration has been given almost exclusively to the average reactions of the average subject. While average reactions are the most significant in interpreting what may be expected for the average person for the average exposure, they must be used with caution when determining the worst conditions a person may work under or the length of time a person may work in any hot atmosphere. For this reason the broken-line curves in Fig. 22 were drawn to represent the maximum reactions expected under most circumstances for all workers who have been found to react favorably to hot conditions. This favorable reaction might be brought

about either through acclimatization or adjustment in their physical make-up after experience, or through their having a sufficiently adaptable physical make-up. For each length of exposure these curves were drawn so as to be above most of the individual test points for the different subjects, similar to the

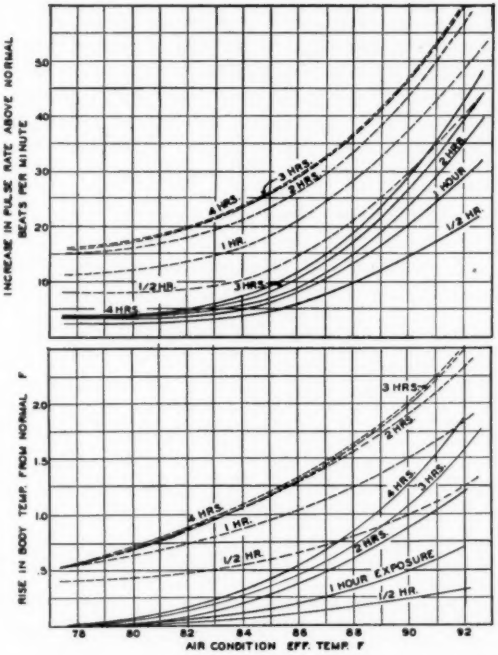

FIG. 22. RELATION BETWEEN THE EFFECTIVE TEMPERATURE OF THE ATMOSPHERIC CONDITION AND THE AVERAGE AND MAXIMUM EXPECTED RISE IN BODY TEMPERATURE AND INCREASE IN PULSE BEAT ABOVE NORMAL FOR ½, 1, 2, 3 AND 4-HOUR EXPOSURES. SOLID LINE CURVES REPRESENT THE AVERAGE RISE IN BODY TEMPERATURE AND AVERAGE INCREASE IN PULSE BEATS OF THE AVERAGE SUBJECT. BROKEN LINE CURVES REPRESENT THE MAXIMUM EXPECTED RISE IN BODY TEMPERATURE AND THE MAXIMUM EXPECTED INCREASE IN PULSE BEATS FOR A PERSON PHYSICALLY FIT AND ACCLIMATED FOR WORK IN HOT ATMOSPHERES

broken-line curves marked maximum reactions in the middle sections of Figs. 8 and 11.

In considering the physical data plotted in Figs. 7 to 21, it is of interest to compare the intensity of physiological reactions, as found in this study, to the similar reactions for persons *seated at rest* and for persons performing *heavy*

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work, in the earlier Laboratory studies. The metabolic rates for these different conditions have already been compared in Fig. 17, while the weight loss for the three types of activities was presented in Fig. 18. The relation between rise in body temperature and increase in pulse rate, and the effective temperature for the three activities are given for comparison in Fig. 23. The curves for *light work* in this figure were taken as the averages for all three humidities in Figs. 8 and 11 of this study, while the curves for *at rest* and *heavy work* were taken from data published in earlier Laboratory reports.[1, 5] In Fig. 24 are given the relations between sensible perspiration on the body and forehead and the effective temperature of the air condition for persons *at rest, light*

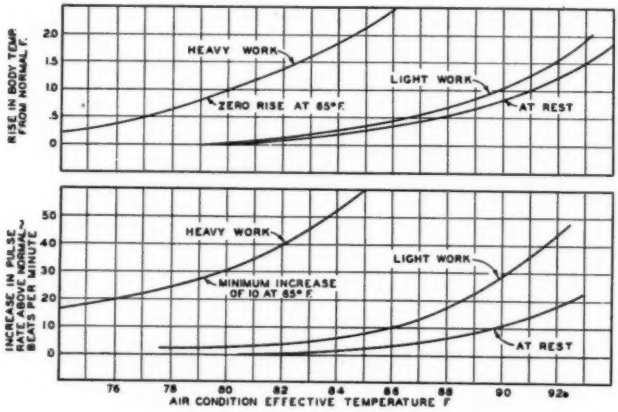

FIG. 23. RELATION BETWEEN INCREASE IN PULSE RATE AND BODY TEMPERATURE FROM NORMAL AFTER 3 HOURS EXPOSURE AND THE EFFECTIVE TEMPERATURE OF THE AIR CONDITION IN THE TEST ROOM FOR PERSONS AT REST, AT LIGHT WORK (THIS STUDY) AND AT HEAVY WORK

work, and *heavy work*. The *light work* curves were taken directly from Figs. 19 and 20 of this report, while the data for *at rest* and *heavy work* were again taken from earlier Laboratory reports.[1, 5]

It is hard to account for the practically equal rise in body temperature and the greater degree of sensible perspiration at a given effective temperature for persons seated *at rest* when compared with persons standing and performing *light work* as in this study. Since these former data were collected several years ago under different conditions, it is possible that the interpretation of sensible perspiration might have differed slightly. On the other hand, these facts may be accounted for by the purely physical factor of increased rate of heat dissipation for a person standing up and moving about, as discussed previously.

The purpose of this study is to supply information which may be used in the

guidance of labor, employers, and others interested in the economic and social aspects of working conditions in determining atmospheric conditions under which a person may work a full day or shorter periods without harmful effects. The curves for rise in body temperature and increase in pulse rate for different times of exposure, shown in Fig. 22, offer a ready picture to enable one to determine the relation between time of exposure and the effective temperature, for these physiological reactions.

This study does not in itself offer conclusive information on the severity of a physiological reaction that a worker may be expected to endure without harmful effects on his health, or without undue discomfort. Comparison of the data plotted in Figs. 8 and 11 indicates that an increase in pulse rate of about 24 beats per minute accompanies a rise in body temperature of about

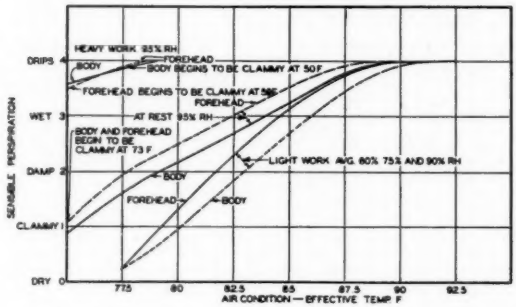

FIG. 24. RELATION BETWEEN SENSIBLE PERSPIRATION ON THE BODY AND FOREHEAD AND THE EFFECTIVE TEMPERATURE OF THE AIR CONDITION IN THE TEST ROOM FOR PERSONS AT REST, LIGHT WORK (THIS STUDY) AND HEAVY WORK

1 deg, for moderate conditions of exposure. For more severe conditions the pulse rate increases proportionately at a higher rate for the same rise in body temperature. Rises in body temperature of ½, 1, 1½ and 2 deg are, respectively, accompanied by increases in pulse rate of 12, 25, 38 and 70. The fact that rise in body temperature and increase in pulse rate always appear to be a positive index of exposure to hot conditions makes it seem most desirable to base any limitation, either on length of exposure or severity of a hot condition in which a man should work, on these two factors. As pointed out previously, rise in body temperature represents a more positive mathematical relationship to the atmospheric condition, and therefore, it should be accepted as the better index of the two.

While this study was not designed particularly to determine the allowable rise in body temperature of workers in industry, the observations of the effect of temperature rise on subjects in this study as observed by the investigators and the medical men in attendance offer some basis for a limitation. These observations have led the authors to recommend that the limitations be so

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TABLE 3. PHYSIOLOGICAL CHANGES AND CONDITIONS ACCOMPANYING A MAXIMUM EXPECTED RISE IN BODY TEMPERATURE OF 1½ DEG AND 1 DEG IN FROM ½ TO 4 HOURS EXPOSURE. ALSO SIMILAR DATA ACCOMPANYING A ½ DEG AVERAGE RISE IN BODY TEMPERATURE IN EQUAL EXPOSURES

LENGTH OF EXPOSURE	E. T.–F	MAX. EXPECTED		AVERAGE		VITAL CAPACITY		LEUCOCYTE COUNT		WEIGHT LOSS		PERSPIRATION				COMFORT	
		BODY TEMP. RISE–F	PULSE INCREASE BEATS/MIN	BODY TEMP. RISE F	PULSE INCREASE BEATS/MIN	AVG. % CHANGE	MAX. EXPECT. % CHG.	AVG. PER CU. MM	MAX. PER CU. MM	CHANGE OZ./HR. AVG.	CHANGE OZ./HR. MAX.	BODY AVG.	BODY MAX.	FOREHEAD AVG.	FOREHEAD MAX.		
1/2	93.5	1.5	49	0.38	25												
1	90.0	1.5	37	0.50	20												
2	87.6	1.5	32	0.55	13												
3	87.4	1.5	34	0.62	15	-2.8	-8.2	8700	9600	6.3	13	WET	RUNS	RUNS	RUNS	VERY WARM	VERY WARM
4	87.2	1.5	33	0.70	15												
1/2	90.0	1.0	30	0.25	14												
1	85.6	1.0	20	0.18	7												
2	83.4	1.0	20	0.16	6												
3	83.3	1.0	22	0.21	6	-1.5	-6.5	8200	8900	4.5	9	DAMP	WET	WET	RUNS	COMF	VERY WARM
4	83.1	1.0	22	0.23	6												
1/2	95.0	1.75	57	0.50	30												
1	90.0	1.50	37	0.50	20												
2	87.1	1.43	30	0.50	12												
3	86.4	1.35	30	0.50	11	-2.3	-8.0	8600	9400	5.5	11.5	WET	RUNS	RUNS	RUNS	VERY WARM	VERY HOT
4	85.6	1.27	28	0.50	11												

placed that no worker be expected to have more than 1½ deg rise in body temperature. Table 3 gives some of the maximum expected and average physiological reactions which would accompany a 1½ deg maximum rise in body temperature. A lack of proportionality between the maximum and average rise in body temperature is apparent and may be due to greater errors in locating the maximum curves from test points. It will be noted that the different lengths of exposure in the several effective temperature conditions all give an approximate average rise in body temperature of ½ deg. If a ½ deg average rise were accepted instead of a 1½ maximum rise, atmospheric conditions of 95, 90, 87.1, 85.6 and 86.4 deg ET could be endured for ½, 1, 2, 3 and 4 hours, respectively.

The major criticism of this investigation as outlined and performed has been that the study would give data only which might be used in limiting hot atmospheric conditions for one type of physical activity, and further, that this particular activity does not accurately simulate any practical work in any industry. A valid answer to this criticism is found in the often demonstrated fact that metabolism is a true and acceptable (in fact, the only true and acceptable) index of intensity of physical activity or work. Hence, all types of work resulting in a given metabolic rate in the same atmospheric condition will undoubtedly result in the same heat elimination problems and therefore the same rise in body temperature, increased pulse rate, and other physiological reactions resulting from interference with heat dissipation from the body.

If this point of view can be accepted, and it is believed that it can, then the relation between physiological reactions and the length of exposure to the effective temperature of the air condition for a relatively few rates of physical activity should make it possible to determine these relationships for any other rates of work. The establishment of such relationships is attempted in Fig. 25, where rise in body temperature and increase in pulse rate for various lengths of exposure are plotted against the metabolic rate. This study offers rather complete information for these relationships for *light work*, accompanied by a metabolic rate of 76 cal per square meter per hour, and earlier Laboratory studies [4] offer a limited amount of information for persons *at rest* and at *heavy work*. Based upon the average rise in body temperature and increase in pulse rate during a 3-hour period, the same solid-line curves are drawn through well-established points for the three metabolic rates. Maximum expected rates of rise in body temperature and increase in pulse rate for a 3-hour period are available both from this study and the earlier investigation for persons *at rest*, and it was these data and the position of the solid-line curves on this figure that were the determining factors in locating the broken-line curves.

While the meager amount of data available for plotting the curves in Fig. 25 does not establish too accurately their exact position between *light work* and *heavy work*, particularly when the maximum physiological reactions are considered, it does indicate that with relatively little added information curves acceptable for practical use could readily be established.

In order to supply sufficient information for accurately plotting curves similar to those in Fig. 25, another series of tests should be made with the subjects performing *heavy work*, so as to establish with greater confidence

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these latter points. In addition, a somewhat less comprehensive study at a metabolic rate somewhere in the neighborhood of 100 cal per square meter per hour should aid in fixing the curvature and exact location of the curves

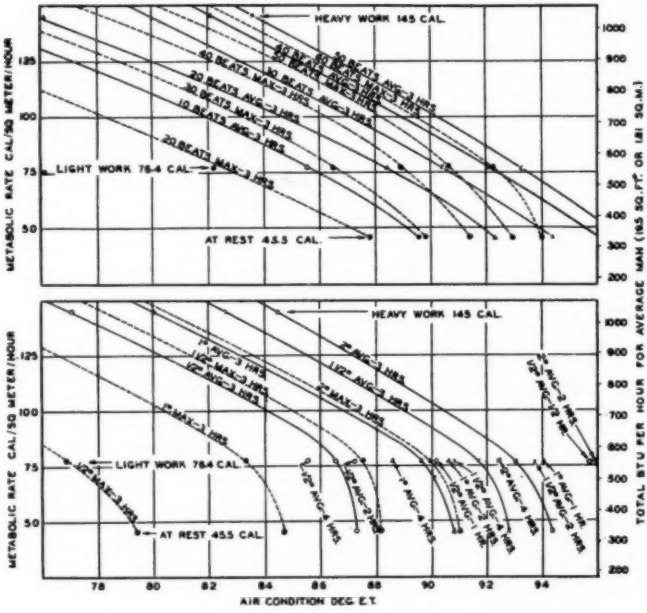

HEAVY WORK 145 CAL.
LIGHT WORK 78.4 CAL.
AT REST 48.5 CAL.
METABOLIC RATE CAL/SQ METER/HOUR
TOTAL BTU PER HOUR FOR AVERAGE MAN (185 SQ.FT. OR 181 SQ.M.)
AIR CONDITION DEG. E.T.

FIG. 25. RELATION BETWEEN THE EFFECTIVE TEMPERATURE OF THE ATMOSPHERIC CONDITION AND THE METABOLIC RATE FOR RISE IN TEMPERATURE OF ½, 1, 1½ AND 2 DEG, AND FOR INCREASE IN PULSE BEAT OF 10, 20, 30, 40 AND 50 BEATS PER MINUTE. SOLID LINE CURVES REPRESENT THE AVERAGE RISE IN BODY TEMPERATURE AND THE AVERAGE INCREASE IN PULSE BEATS OF THE AVERAGE SUBJECT. BROKEN LINE CURVES REPRESENT THE MAXIMUM EXPECTED RISE IN BODY TEMPERATURE AND MAXIMUM EXPECTED INCREASE IN PULSE BEATS FOR A PERSON PHYSICALLY FIT AND ACCLIMATED FOR WORK IN HOT ATMOSPHERES. MAXIMUM CURVES DRAWN THROUGH POINTS INDICATING METABOLIC RATES OBTAINED FROM MAXIMUM AT REST AND LIGHT WORK DATA ARE DRAWN SIMILARLY TO THE CURVES FOR THE AVERAGE REACTION. POINTS INDICATING THE POSITION OF CURVES FOR SEVERAL RISES IN BODY TEMPERATURE IN FROM ½ TO 4 HOURS GIVEN FOR LIGHT WORK ONLY

between *light work* and *heavy work*. This seems particularly desirable because of the apparent rapid change in the direction of these curves in the neighborhood of *light work*. Moreover, since this study was limited to high humidities, a few additional tests should be made at lower humidities.

With sufficient additional data and the development of perfected charts similar to Fig. 25, one should be able to determine with confidence the temperature rise and the pulse increase for any period of exposure in any hot atmosphere for all rates of work for which the metabolism is known. While the amount of work incident to a continuation of this study is rather extensive, it is quite insignificant when compared with the consequential effects it should have in the application of air conditioning to industry. The cost should be insignificant when compared with the economic saving in health, efficiency, comfort, and the satisfaction accruing to labor, industry and the entire social system.

It is of interest to consider a hypothetical application of the information contained in Fig. 25, to an unstudied type of employment, assuming that no change in the character of the curves in Fig. 25 would result from further study. A few workers performing the new type of work in question would be given a metabolism determination while working. Assume that they show a metabolic rate of 100 cal per square meter per hour, and that the limitation of 1½ deg maximum rise in body temperature suggested is to govern the case. The broken-line curve, 1½ *deg max* 3-*hour*, will apply, which shows that the worker can engage in this new activity for 3 hours at about 85 deg ET. Similar charts supplied for ½-, 2- and 4-hour exposures would make it as simple to determine the severity of the atmospheric condition which would give a maximum expected elevation of body temperature of 1½ deg in these lengths of exposure.

In order to make available adequate data for estimating the minimum air conditioning requirements for hot industries, future work of the Committee should therefore include:

1. Additional tests with the same type of *light work* in the same atmospheric conditions during the winter season, in order to determine whether men working under these conditions react the same in winter and summer.

2. A series of tests with the same type of *light work* at a lower relative humidity, for example 30 per cent, in order to more adequately determine whether effective temperature alone need be considered, when a wider range of relative humidities is encountered.

3. A series of tests with *heavy work*, in order to supply adequate data for work giving a metabolic rate of 145 cal per square meter per hour, so as to more satisfactorily locate the curves for average and maximum reactions as drawn in Fig. 25.

4. A series of tests with a rate of work giving a metabolic rate of about 100 cal per square meter per hour, in order to better establish the curves drawn in Fig. 25, between *light* and *heavy work*.

5. A series of tests at rest, collecting data under precisely the same atmospheric conditions as in this study, in order to verify or correct the apparent conclusion from this study that a person standing up and performing *light work* has about the same rise in body temperature, a somewhat less degree of perspiration, and less feeling of warmth than a person seated at rest.

6. Additional work to determine the effect of air movement in reducing the severity of the physiological reactions for different types of work at the same dry-bulb temperature and relative humidity. In this connection, work already carried on at the Laboratory,[5, 9, 10] has established the effect of air movement on the comfort and presumably the physiological reactions of persons at rest, while a smaller amount of data has been reported for persons working. This new phase of the study should be so planned as to determine the extent to which these earlier findings may apply.

CODES

Within the next few years, there may be anticipated the enactment of many laws and regulations of a social nature. No doubt many of these will have for their subject air hygiene as affecting the health of the worker. This work under the Committee on Air Conditioning in Industry should offer a true foundation for these codes and standards. It may be assumed that regulatory bodies are responsible only for the health of the workers and are therefore interested in those conditions of the atmosphere which adversely affect it. As an example for the need of an understanding and knowledge for those conditions of temperature and humidity which may affect the health of the worker, the State of Massachusetts has had for several years regulations limiting temperatures and relative humidities in textile factories. There can be no doubt that those responsible for these regulations were moved only for the best interests of the worker and that the limiting atmospheric conditions written into the regulations were for his protection.

The maximum dry-bulb temperatures and absolute humidities specified in the Regulations, interpreted as effective temperatures, range from a maximum of 89.3 deg ET to a minimum of 59.3 deg ET. Assuming for the moment that the work of this summer was more than of an exploratory nature and that the critical effective temperature for health occurred when there was a maximum expected increase for any worker of 1.5 F in body temperature with an increase of pulse of 28 beats per minute; or an average increase for the group of 0.5 F in body temperature and 12 pulse beats per minute, then the conditions laid down in the Regulations show a need for fundamental data. All those specified conditions above 91 F dry-bulb are beyond the critical temperature as found for this intensity of activity, so the purpose of the Regulations is not fulfilled as far as the health of the worker is concerned. Those temperature and humidity conditions below 91 F in the Regulations have no effect on the health of the worker and are only a manufacturing limitation. This code was copied from the English Act of 1911 which has been revised in 1935. The revised English Act limits the operation of the mill to times when the wet-bulb temperature does not exceed 80 F.

Codifying critical effective temperatures for industries of the textile mill type is rather simple when compared to those which have conditions above the critical effective temperature during a large portion of the time. If the same criteria of health condition is maintained, it may be necessary and advisable to make provision in the codes for a physical examination of the worker, thus eliminating the unfit and also to provide a limitation of the duration of continuous activity between rest periods. The temperature and humidity found in deep mines offer an excellent example of these extreme conditions. The general sources of heat in the mines are wall surfaces, men, explosives and the cooling of freshly opened rock. The heat from the freshly opened rock is by far the most important. Many of the deep mines are naturally wet and added to this natural moisture in those where the rock is of a siliceous nature, water is used to settle the dust with a consequent high relative humidity. Saturated temperatures of 90 to 95 F are frequently encountered. In the mining district of South Africa, a temperature of 93 F saturated is considered the critical above which many cases of heat prostration occur.

By government regulation, 30 cfm of air must be supplied per person

employed underground throughout the 24 hours. Working periods are short, interspersed with rest periods. In other places, many of the deep mines are dry with a much higher rock temperature. The resulting effective temperature is, however, approximately the same and the problem of protecting the health of the worker remains.

When codes are written for industrial conditions of this severity, the laboratory findings of the physiological effects on the worker of these intense conditions of temperature and humidity will be of great value and will supply necessary fundamental information on which to base a code protecting the health of the worker and benefiting the employer through a smaller labor turnover and a greater efficiency of labor.

ENGINEERING

This research will, when completed, determine the critical effective temperature for persons engaged in various degrees of activity. It is expected that as a result of this research tabular information may be compiled giving the approximate critical effective temperatures corresponding to different activities within certain predetermined limits of rise in body temperature and pulse rate. These data will act as a guide to the engineer in the selection of air handling equipment necessary to produce atmospheric conditions below the established critical temperatures for different industries.

Because of the wide range of metabolic activities for different types of work it is conceivable that there might be three different general types of equipment used for producing the desired results which may be classified:

1*a*. Ventilating equipment handling untreated outside air.
b. Open windows in side walls and roof.
2. Air conditioning equipment with evaporative cooling.
3. Air conditioning equipment with refrigeration.

Each one of these classes has very definite limitations which depend not only on the activity or rate of metabolism of the persons engaged but also on the atmospheric conditions met with in the different localities. It can be readily seen that critical temperatures approaching even the higher brackets may be found in subtropical climates and it is also true that in some localities in the north and located on a body of cold water, conditions of quite low critical temperatures could be met without any equipment or that handling only untreated outside air.

In Fig. 26 will be noted three critical effective temperatures corresponding to a 1½ F body temperature rise that have been selected from data reported in this paper and which are enumerated herewith:

	CRITICAL, EFFECTIVE TEMP DEG	METABOLIC RATE, BTU PER HOUR
1. Persons at rest	92.8	384
2. Light Metal Worker	86.5	862
3. Stone Mason	72.8	1,490

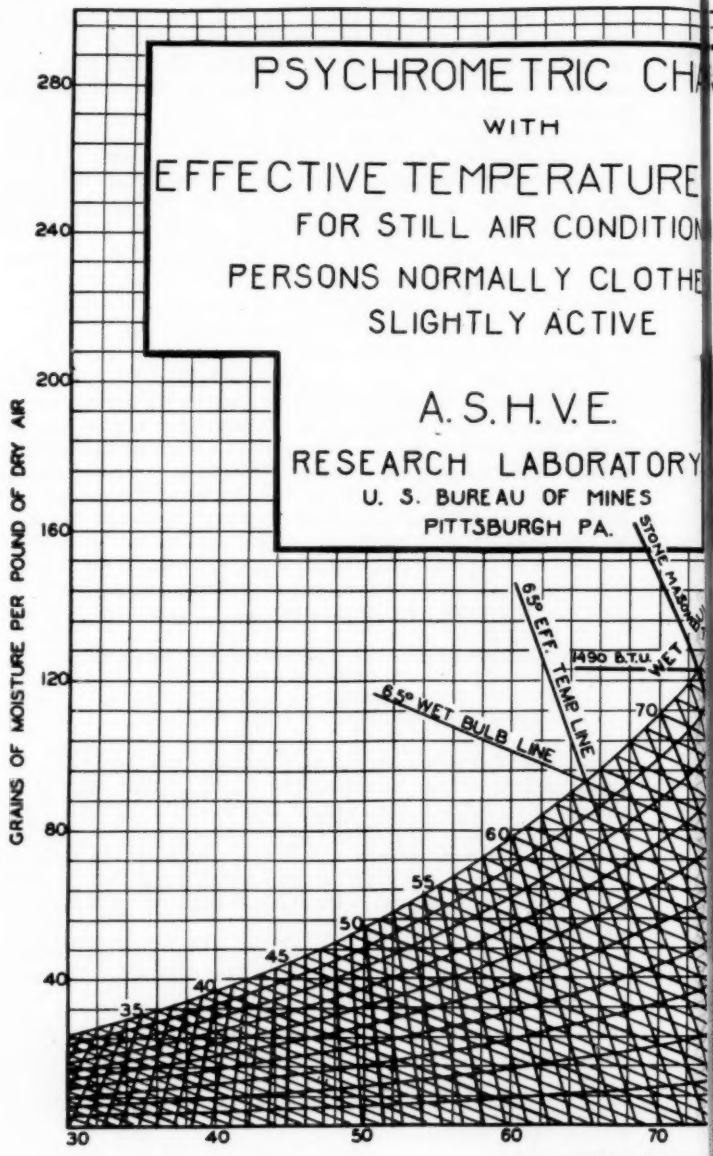
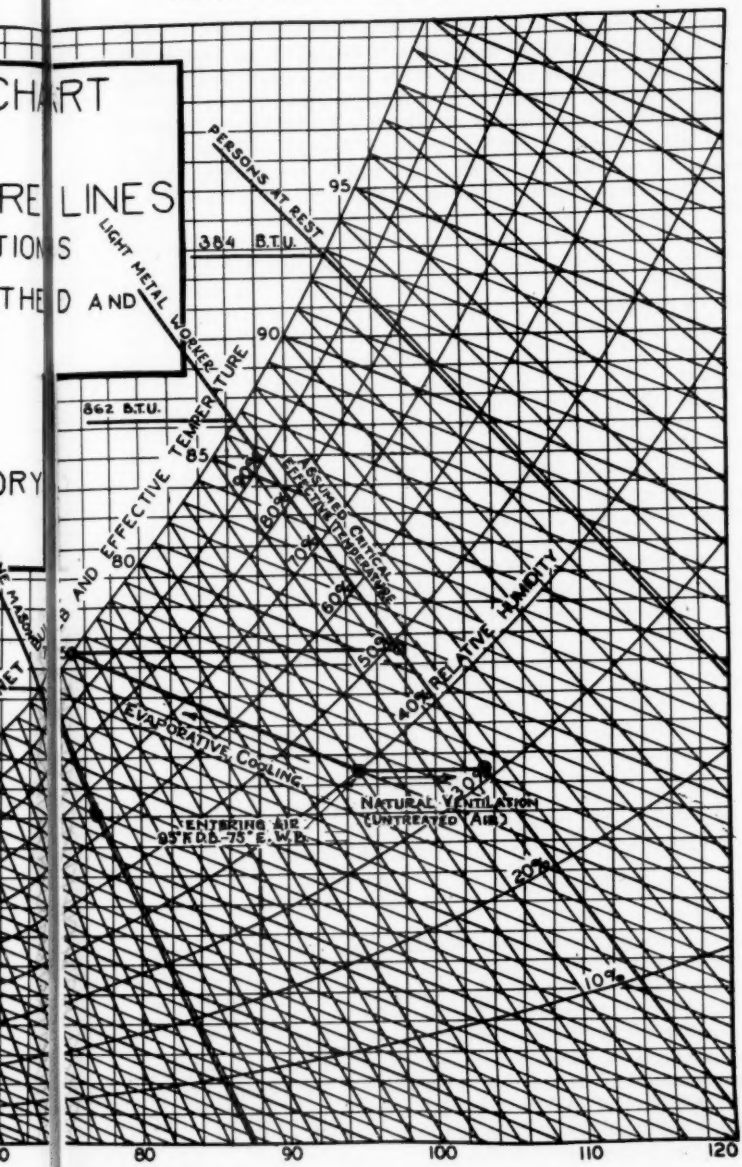


FIG. 26. PSYCHROMETRIC CHART SHOWING CRITICAL EFFECTIVE
PROCESSES TO PRO-



TEM-
ECTIVE
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TEMPERATURES FOR THREE ACTIVITIES AND AIR CONDITIONING DUCE DESIRED RESULTS

Also on this chart is plotted the maximum outside air condition of 95 F dry-bulb and 75 F wet-bulb which is assumed in this case to be applied for illustration to the critical effective temperature line of 86.5 deg corresponding to a metabolic rate of 862 Btu per hour of a light metal worker.

With these assumed conditions it would be necessary for equipment handling untreated outside air to have a capacity in cubic feet of air per minute sufficient to absorb the sensible heat in the room with a temperature rise of 8.8 F which would result in a final condition of the work space of 103.8 F dry-bulb and 77.3 F wet-bulb as may be referred to in Fig. 26.

In those cases where evaporative cooling may be used the amount of cooling depends not only on the outside wet-bulb temperature, but also on the maximum humidity which may be permitted within the enclosure. The amount of air which would be handled by an evaporative cooling device varies directly with the cooling to be accomplished. When evaporative cooling is used for the outside air conditions as previously cited, the capacity of the equipment in cubic feet of air per minute would be based on a temperature rise of 22.8 F instead of 8.8 F when using untreated outside air to obtain the same effective temperature. The resulting condition in the space would be 97.8 F dry-bulb and 80.7 F wet-bulb. Thus it is evident that this type of equipment could be used to produce the conditions below the desired critical effective temperature for the degree of activity assumed. For other outside wet-bulb temperatures the limitations of this type of equipment may be readily visualized on the chart. As the wet-bulb temperature is reduced, the effective range of temperature of the air introduced is increased which results in smaller air conditioning equipment and the reverse is true with higher initial outside wet-bulb temperatures. It may generally be assumed that with evaporative cooling the effective temperature varies inversely with the relative humidity. For example, if it were possible to maintain within an enclosure 80 per cent relative humidity there would be a temperature rise in the air from a saturation point of 7.5 F dry-bulb. However, if it were possible to maintain 70 per cent in the same enclosure the temperature rise of the air would be 11.5 F so that the amount of air required to absorb any specified amount of heat would be proportionately reduced. Unfortunately, the maximum cooling to be obtained by this method is limited by the outside wet-bulb temperature. If a locality had an outside wet-bulb of 75 F and it were possible to carry 70 per cent relative humidity within a plant, then the inside temperature of the plant would be 86 F which is, of course, based upon providing sufficient air to absorb the heat load produced within the plant. These conditions correspond to an effective temperature of 81.2 deg.

In many industries, 65 per cent relative humidity is considered the maximum allowable due to danger from corrosion, etc. With 65 per cent humidity, there is a temperature difference between the saturation point and the dry-bulb temperature of 13.5 F, so that there would result a final dry-bulb temperature of 88.5 F corresponding to an effective temperature of 82.5 F.

As the maximum outside wet-bulb temperature is decreased, so the indoor conditions can be reduced and the reverse is also true. For example, in the south, where it is not unusual to have wet-bulb temperatures of 81 F, then with 65 per cent relative humidity there would be a dry-bulb of 94.5 F corresponding to an effective temperature of 87.3.

To maintain lower critical temperatures it will be necessary to use refrigera-

tion in some form. In the northern part of the country well water of sufficiently low temperature may be obtained. However, in many places it would be necessary to make use of artificial refrigeration. When meeting this condition it might be better from an economic standpoint to shorten the hours of labor or intersperse the periods of activity with intervals of rest so that the same physiological reactions on those persons engaged in this type of work would remain within the healthful limits.

For those degrees of activity where the critical effective temperature is below the outside wet-bulb then some form of refrigeration will be necessary. The capacity of the equipment to fulfill these conditions will depend upon the relative humidity to be maintained within the work spaces. The size of the refrigerating equipment will be smaller as the per cent of relative humidity allowed in the work space is increased, and the amount of air to be handled by the equipment will increase directly with the relative humidity. Assuming that the critical effective temperature for a stone mason is 72.8 deg and that the outside temperature condition is 95 F dry-bulb and 75 F wet-bulb and that it is desired to maintain 65 per cent relative humidity in the work room, it will be necessary to introduce air having a dew-point of 64 F. Therefore, all the air handled by the air conditioning equipment would have to be cooled to 64 F and the amount of air to absorb the heat in the work space would be based on a temperature rise of 13 F which would result in a final condition in the work space of 77 F dry-bulb and 68 F wet-bulb which is equivalent to an effective temperature of 73 deg. The refrigerating capacity necessary to accomplish these results would vary with the requirements for ventilation. When exhaust systems for removing the dust from the cutting surfaces are such that all outside air is necessary, then the capacity of the refrigerating equipment will be sufficient to lower the total amount of air from 95 F dry-bulb and 75 F wet-bulb to 64 F. If the amount of air which is supplied to maintain the desired critical effective temperature is more than that exhausted, then a portion of the air may be returned to the equipment at room conditions with a decrease in the refrigerating requirements.

Engineering calculations to determine the capacity of the air conditioning equipment may be based on procedure outlined in the HEATING, VENTILATING, AIR CONDITIONING GUIDE, 1938 (Chapters 5, 6, 7 and 8). All sources of heat in the space to be cooled must be carefully surveyed. After the proper amount of air has been determined, it should be distributed in the same pattern as the heat load occurs.

SUMMARY AND CONCLUSIONS

1. The paper reviews the developments of present understanding, or lack of understanding, of the relation between length of exposures of workers to hot atmospheres and the resultant disturbance of their physiological equilibrium.

2. The relations are given between lengths of exposure in hot atmospheric conditions of different effective temperatures and changes in various physiological reactions, such as body temperature, pulse rate, vital capacity, leucocyte count, weight loss, forehead and body perspiration, and feeling of warmth. Body temperature and pulse reactions are indicated as being the most significant reactions correlated with the severity of the hot condition. Moreover, since rise in body temperature is shown to be an adequate index of disturbed physiological equilibrium, it is recommended for use

in limiting the length of exposure of workers in hot atmospheres. An average body temperature rise of ½ deg in the average worker qualified by physical make-up and experience to work in hot industries, or a 1½ deg rise in an occasional worker, is suggested as the limiting reaction. Also given are other physiological derangements accompanying a ½ deg rise in the body temperature of the average worker, or a 1½ deg maximum expected rise in the temperature of the occasional worker.

3. A chart is presented for indicating the rise in temperature or increase in pulse rate for different lengths of exposure in atmospheres of different effective temperatures, for different activities, expressed by their equivalent metabolic rate. It is shown that a relatively small amount of additional data to be collected in similar studies covering the necessary range in temperatures, humidities and air motion conditions will be adequate for interpreting the results over the entire range of possible human work. A program for research to supply this information is outlined.

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DISCUSSION

DR. THOMAS BEDFORD[1] (WRITTEN): It is stated in this paper that all types of work resulting in a given metabolic rate in the same atmospheric condition will undoubtedly result in the same heat elimination problems. I do not think this statement is strictly accurate, inasmuch as tasks which involve the same metabolic rate may be performed with varying mechanical efficiencies, and thus the proportion of the total energy production which must be dissipated as heat will vary.

My other point is one of historical interest. You refer to what Cardinal Polignac is alleged to have said in 1713. I do not know whether you took this reference from the report of the N. Y. State Commission or not, but I think it was Bernan (1845) who first attributed this statement to Cardinal Polignac. Actually, the statement appears in a book written under the name of Nicholas Gauger. Bernan, for some

[1] London School of Hygiene and Tropical Medicine, University of London, W.C. 1, London, England.

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reason thought Gauger was a pseudonym adopted by Cardinal Polignac, but in the Biographie Universelle, 16th Volume, published in 1816, there is a note about Nicholas Gauger. Apparently he was a real person. There is an interesting and amusing point about this. In 1850, five years after Bernan's great book appeared, a man named Tomlinson, of London, published a popular book on warming and ventilation. He was a barefaced plagiarist, it would seem, and unblushingly used much of the material in Bernan's book, including all his references to what "the Cardinal" said and did. Some years later, in a further edition, Tomlinson had a very bitter note on Bernan's error in leading him and others astray. He even counted up the number of times Bernan referred to "the Cardinal," but he made no apology for having made such free use of Bernan's work without due acknowledgment. I am greatly interested in the good work you people in Pittsburgh are doing, and look forward to seeing your further results.

W. H. CARRIER: In these tests on critical effective temperatures, the question of clothing is important. It makes a great difference, apparently, because in the South African mines they are working with an 88 or 89 per cent saturation and a wet-bulb of 94 deg which you see would be a much higher effective temperature, and they are really doing work down there, but they are stripped absolutely except for a loincloth.

I think it perhaps would be well to draw attention to the effect of clothing because we might have some conditions which, if tried in South Africa, would result in their not having anybody working in those mines. They do get heat strokes at 94 deg wet-bulb, but at 88 deg, which they reduced it to, they have no heat strokes, 88 deg wet-bulb with about 88 or 89 per cent saturation.

COL. W. A. DANIELSON: Did they take into account the saline content of the blood?

PROF. C.-E. A. WINSLOW: I would like to know whether the leucocyte count was affected at all by changes in blood volume, which I suppose is the thing Colonel Danielson is thinking about too, but I mainly want to pay a tribute to this very remarkable piece of work. I more than agree with what has been said. This is one of the most important things the Society has done to contribute to the art since the original work in the Research Laboratory, and considering the time and the limitations I think it is a most extraordinary contribution.

W. B. HODGE: My experience has been largely with the textile industry, and we are having very high temperatures, as has been brought out, but I would like to ask the writers whether any consideration has been given to these textile operators that work longer than three or three and a half hours a day. Where we have eight hours a day and 44 hours a week, with two shifts, sometimes three, what is the cumulative fatigue beyond the three and a half hour limit? I would be interested to know something about that.

W. L. FLEISHER: I would like to remind the authors of a question I thought should be brought up, as to the difference in reaction due to the skin pigmentation of the colored worker and the other worker.

One more question which I thought might come up is the fact that the comfort sensation of these workers was about 78 ET as against our established comfort condition of 72. I would like, if possible, an explanation of that.

PROF. JAMES HOLT: I would like to ask one question in connection with the textile industry. I have noticed that a great many of the New England textile workers who have gone to the South, and consequently undergone a change of environmental condition under which they normally live, have found it impossible to work in the southern mills and eventually they come back to New England. Is there anything

in the work that is being done that would indicate the effect of environmental conditions upon these workers?

E. HOLT GURNEY: Is there any prospect of work being done of an experimental nature this coming year at the higher metabolic rates?

LIEUT. COMDR. R. M. WATT, JR.:[2] I should like to ask the authors if, in this work, they have developed any information which would bear on a problem which occurs in my work, which is the ventilation of warships. We frequently have engine rooms and spaces where the heat loads are so high there is no possibility whatsoever of bringing down the effective temperature to reasonable limits, and we are attempting to solve the problem by cold blasts of air at the watch stations where the men will be on duty. In your work I should be interested to hear if you have developed any ideas of how effective that is and what we might expect in the way of effective results from such cooling rather than general air conditioning.

F. C. HOUGHTEN: As pointed out by Mr. Bedford, I agree that efficiency of the worker is a factor at least in theory. In estimating heating and cooling loads, we, of course, recognize this as an important point when dealing with electric motors performing work on machines, when either the motor or the work producing machine, but not both, is in the space to be cooled. Some thought was given to the parallel application of a person working in hot atmospheres. It was assumed however that, first, a human being at work is always very inefficient and that most of the mechanical work, and therefore most of the energy transformed to heat takes place within the body rather than appearing as useful work transformed into heat external; second, it was assumed that there probably was not a very large variation in the low efficiency for various work. There are only a few determinations available which allow an estimate of human efficiency. In these cases the useful work performed constitutes only from $\frac{1}{10}$ to $\frac{1}{20}$ of the metabolic rate. Hence, while this is a theoretical factor, it is felt that it is not of great moment.

Mr. Bedford's references to the historical development with additional bibliography are a valuable addition to the subject.

In reference to Fig. 25 in the paper, Dr. Ferderber pointed out that it is a composite chart into which we hope to picture the result of all of this study, and probably several additional studies. In this chart is plotted the metabolic rate against the effective temperature of the air condition. This particular study was made at a metabolic rate of about 75 calories, actually 76, and the data at that level on the chart are satisfactory.

There are a few data, resulting from this study, for persons at rest while seated in the preliminary rooms, in addition to other data for persons at rest from other studies made at the Laboratory. There are a few data collected some 10 or 15 years ago by Dr. W. J. McConnell and others at the Society's Laboratory in quite a different study, some of the data of which could be fitted into this picture.

The combinations of work and atmospheric conditions are plotted in the lower part of the chart, which will give what may be considered reasonable limitations in rise in body temperature. As Dr. Ferderber pointed out, these are not assumed to be finally agreed upon. For instance, if a person works at the rate of 76 calories for 3 hours, he may expect to have ½ deg rise in body temperature, an atmospheric condition of about 78 deg ET, and at a higher effective temperature if seated at rest.

We have figured that a rise in body temperature of about 1-½ deg maximum on occasion by the occasional workers is probably a reasonable limitation. That is comparable with a ½ deg rise in body temperature for the average individual on the average occasion.

The incompleteness of this chart comes from the fact that the data are incomplete

[2] Lieutenant Commander, (CC), U. S. Navy, Navy Dept., Washington, D. C.

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for metabolic rate above 76. We have no data there for work between 76 and 145 calories, and the fact that there is a peculiar curvature to the curves in that region of the chart makes the results doubtful. We are not so certain that these data for persons at rest fit exactly the same conditions as the data for light work, and therefore we are not so sure of the curvature, and not having any data between a metabolic rate of 76 and 145, we are only making a guess in saying that these curves as drawn approximate the real curves between those points. But if and when we do complete this picture, a relatively small amount of additional data should be required, and if you once accept a limiting rise in body temperature for any given condition, you can apply it for all workers, and for all activities.

A question was asked: "Will additional work be done at higher rates of work or at higher metabolic output?" Yes, our plans very definitely include such additions to the program. Whether it will be carried out this summer or later, it is not definite, but they certainly should be included in the program.

Mr. Carrier raised the question of clothing. It is true that the less clothing a person wears the hotter the condition he can work in for the same length of time, or the harder he can work in a given condition. This study was presumed to apply in industries where both men and women work. We had specifically in mind its application to the textile industry, but we do not feel it is limited to that. The clothing worn in the study is apparently about the average type of clothing worn in such work; that is, a very light pair of trousers, thin sleeveless undershirt, shoes and socks.

The question of salt was brought up, and, of course, that same question is also involved in Professor Winslow's question of the blood volume. There have been many suggestions that should have been included in this study, a study of salt, that is, the salt content of the blood and the body. If the study had been made 10 or 15 years ago, that certainly would have been the case, but I do not know whether you know it or not but the very comprehensive work that has been done on the need of salt replacement in the bodies of workers who perspire greatly was in some measure initiated and furthered by the work at your own Research Laboratory when Dr. McConnell and others were working on related subjects in the early twenties.

In this study we only saw to it that these men had a normal intake of salt, so as to not deplete their salt content.

Now, the particular application to Professor Winslow's question of blood volume. Of course, if you perspire you deplete the water content of the body and of the blood. These men were allowed to drink whatever water they wanted. You notice we plot the weight loss. The weight loss was determined by any actual loss plus any water intake during the period less any urine loss; in other words, it was the net loss by perspiration. In very few cases was there actual decrease as loss in weight during a period of work, which means that the water intake was equal to the perspiration loss.

Mr. Hodge asked concerning the accumulative effect of fatigue beyond three hours. These studies were continued in most cases to 4 hours excepting in such cases where the condition was so severe that the men had to quit in 3 hours. The reason we based our plots mostly on 3 hours was the fact that we could include those tests in which the time period was limited to 3 hours because of the study. Fatigue, of course, is accumulative and probably increases more than in a linear relation to time, but these curves can be extrapolated to longer periods of time. Four hours was taken as a reasonable length of time, since it represents rather definitely a single shift in a work day which is followed by a rest period.

Mr. Fleisher asked regarding relative reactions of the colored man and the white subjects. There was no difference. It should be remembered that there were very few tests made on this colored man. Most of the tests were made on the college students who were more easily available and who could be relied upon for taking,

recording and analyzing some of their own data, but we did enough work on other subjects to show that we got the same reactions. There were no measurable differences in reactions with age of the subject or with color.

A very important question in connection with this study is the peculiar fact shown by the plotted data that these men were comfortable at 78. The data accumulated were based upon our accepted scale of comfort, but when we came to analyze the data, much to our surprise we found that these fellows said they were comfortable on an average at about 78 deg ET rather than 71 to 73 F found in other Laboratory studies for persons seated at rest in an audience hall, or 68 and 69 as will be shown working at an operating table. I am not sure what the significance of this difference is, but I believe it is due to an interpretation on the part of these men of when they were comfortable. After all, that is a very important question in any comfort study; that is, to know just what a man means when he says he is comfortable. I believe that these men meant by the fact that they were comfortable, that they were not uncomfortable, and I think if you could analyze their sensations you would find that they did not consider the same level of comfort that a person would if seated at rest in a room of this kind.

Professor Holt asked concerning the acclimatization of men going from the north to the south. I do not know how to answer that. It is true, of course, that southerners have a different metabolism. I think their pulse rate is known to change after living in the south. Whether that is true or not I do not know.

There was a question asked concerning the application of the data to conditions on a warship, presumably in a boiler room or other places below where it is very hot. I do not think there is any question but that more complete data of that type including data for men at hard labor will apply. First of all, and this gives a good opportunity to bring out how you would apply this chart, assuming that this chart was complete and we could, with confidence, accept all of the data plotted on it, the first thing required would be to determine the metabolic rate of these men in a boiler room or below on the warship. If they were working at such a rate as to give the metabolic rate of our subjects in these studies or, in other words, about 76 calories, then it could be determined directly from the experimental data in this paper.

I think that covers pretty well the questions asked. Were there any others that I have not covered?

PROF. F. B. ROWLEY: I should like to ask whether in making up the figures for the curves the temperatures of the surrounding objects were considered. It would appear that in some industries in which there were furnaces or other objects of high temperatures the effective air temperatures required would be quite different than those for other industries in which the surrounding objects were of low temperatures.

MR. HOUGHTEN: That is an important question. There are, of course, several factors that determine a person's feeling of warmth. There is the dry-bulb temperature, the wet-bulb temperature, the air velocity and the radiation. These studies were made by eliminating largely the effect of radiation. They were made in our psychrometric rooms in which the walls were well insulated and where therefore the wall temperatures were practically in equilibrium with the air.

Now, if radiation is an important factor, then certainly there will be a different application and that probably will require another correction to this whole chart, as it should require, and does require, a correction or different application of the entire effective temperature idea.

The importance of radiation in any comfort work has come rapidly to the foreground in the past 2 or 3 years and it is sometimes indicated and assumed that in the Society's early work it was forgotten. It was not forgotten.

You will find recurrently in those early Laboratory reports it was definitely pointed out that the work was done in places or in environments where radiation was not

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an important factor. It was not an important factor because we eliminated it in order to eliminate one factor that could be eliminated, and in those earlier studies it was urged that the work should be continued to cover conditions where radiation was an important factor.

So the position and the condition of the whole background of the Society's comfort data is this: it applies where radiation is a minor factor; in other words, where the surrounding walls and objects are essentially at the same temperature as the air. For any other application radiation should be studied.

MR. CARRIER: May I add one thing to that matter of radiation? I think it is pretty difficult, as Mr. Houghten mentioned, to duplicate any radiation because it varies with the areas, and rarely do you have radiation uniformly from all directions. This is one reason for limiting, but if you will take a thermometer with a heat-absorbing cover, you will get a temperature which will agree perfectly with your sensible temperature reading, and that is really, I think you will find from your studies, all that you need and that is about the only way that you can integrate the effect of combined radiation and convection.

No. 1107

RECENT ADVANCES IN PHYSIOLOGICAL KNOWLEDGE AND THEIR BEARING ON VENTILATION PRACTICE

REPORT OF THE ASHVE RESEARCH TECHNICAL ADVISORY COMMITTEE ON PHYSIOLOGICAL REACTIONS: C.-E. A. WINSLOW, *Chairman;* DR. T. BEDFORD, DR. E. F. DUBOIS, DR. R. W. KEETON, A. MISSENARD, DR. R. R. SAYERS, AND C. TASKER

INTRODUCTION

SUBSTANTIAL advances have been made in recent years in the understanding of the fundamental physiological problems underlying the art of air conditioning. It is both unnecessary and impracticable to review here all the researches which have been published in England, France, and Canada, and at the ASHVE Research Laboratory, the U. S. Bureau of Mines, Cornell University, Harvard University, the University of Illinois, the Russell Sage Institute of Pathology, the John B. Pierce Laboratory of Hygiene, and elsewhere in this country, but it does seem worth while to call attention to a few of the salient points which have been brought out and to indicate certain of the immediate implications of these researches for air conditioning practice.

It may safely be said that the course of physiological and clinical studies has consistently tended to emphasize the problem of heat interchange between the body and its environment as of basic and fundamental importance. The search for mysterious chemical and electrical relationships has continued, but to date there is no valid evidence that any such influences are of material significance under ordinary living conditions. All carefully conducted work on the physiological effect of ions has been negative and there is nothing to encourage the wider usage of ionization.

Ozone has been used with success for the destruction of micro-organisms (molds) in meat packing establishments and the like; and where considerable amounts of organic effluvia are present this substance may be useful as a deodorant. For ordinary ventilation practice, however, neither of these purposes can be usefully attained, since the concentration of ozone necessary for effectiveness would be likely to transcend the limit of comfort in ordinary occupied rooms. While ozone has been used in the treatment of certain diseases, there is no evidence that it has a tendency to increase comfort or to

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benefit health under conditions of normal human occupancy. Little or nothing has been added to the old conclusions of the New York State Commission on Ventilation that in the school room the use of ozone would be objectionable rather than desirable. The practice had its advocates for a period in St. Louis, but the expensive apparatus and installation has been completely abandoned.

It should be noted that this report deals with physiological reactions to the air of ordinary occupied rooms and does not consider the removal of industrial

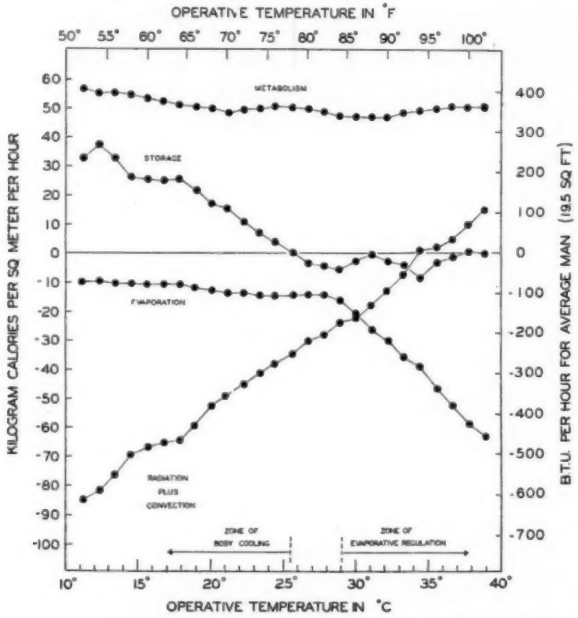
OPERATIVE TEMPERATURE IN °F
50° 55° 60° 65° 70° 75° 80° 85° 90° 95° 100°
KILOGRAM CALORIES PER SQ METER PER HOUR
60 50 40 30 20 10 0 -10 -20 -30 -40 -50 -60 -70 -80 -90 -100
BTU PER HOUR FOR AVERAGE MAN (19.5 SQ FT)
400 300 200 100 0 -100 -200 -300 -400 -500 -600 -700
METABOLISM
STORAGE
EVAPORATION
RADIATION PLUS CONVECTION
ZONE OF BODY COOLING
ZONE OF EVAPORATIVE REGULATION
10° 15° 20° 25° 30° 35° 40°
OPERATIVE TEMPERATURE IN °C

FIG. 1. RELATION BETWEEN METABOLISM, STORAGE, EVAPORATION, RADIATION PLUS CONVECTION, AND OPERATIVE TEMPERATURE FOR THE CLOTHED SUBJECT

dusts or the destruction of germs of communicable diseases in the atmosphere. These problems fall within the scope of other committees.

ZONES OF PHYSIOLOGICAL RESPONSE

So far as physiological reactions in indoor spaces, under conditions of normal human occupancy, are concerned, the basic problems are thermal in nature. Perhaps the first point of importance brought out by recent studies is the fact that the reactions involved in a cold and in a hot environment are radically different in nature. Therefore, any attempt to formulate simple engineering relationships covering the entire thermal scale are obviously doomed

to failure. The most complete information available relates to the lightly clothed subject[1] in a semi-reclining position (John B. Pierce Laboratory of Hygiene). These experiments indicate that the clothed, semi-reclining subject exhibits normal and easy physiological regulation between air temperatures of 77 and 86 F (25 and 30 C). Above 86 F (zone of evaporative regulation) the individual adapts by active sweat secretion. Below 77 F the skin temperature falls, but adaptation is incomplete and the temperature of the body tissues falls more or less continuously (zone of body cooling).

Fig. 1 shows physiological changes in these zones for such clothed, semi-reclining subjects. In the zone of evaporative regulation, metabolism remains constant and increasing evaporative heat loss just balances decreasing convective heat loss and increasing heat gain (above the zero line), due to rising operative temperature. In the zone of body cooling, metabolism rises a trifle, below 68 F (20 C), evaporative heat loss falls very slightly, and storage (cooling of the body tissues) rises sharply as heat loss due to convection and radiation increases.

The conception of *storage* is essential for an understanding of the reactions of the human body to its environment. As used in this report, it implies a rate of cooling (or heating) of body tissues and, according to convention, positive storage involves cooling, and negative storage, heating, of the tissues. This process, if cooling is rapid, cannot be measured by rectal temperature, since a change in the temperature of the circulating blood lags far behind the cooling of the skin and the muscle masses in the limbs and other organs. Furthermore, the process may continue for a long period; in certain recent experiments, both skin temperature and rectal temperature were found to be still falling after a period of six to eight hours on exposure to a cold environment.

REACTIONS IN THE COLD ZONE

An interesting point indicated by the studies at the John B. Pierce Laboratory of Hygiene is that when a given operative temperature (that is, the temperature which should physically exert a certain calorie-demand upon a body of fixed surface temperature) is produced by (*a*) air and walls at approximately the same temperature, (*b*) colder air and warmer walls (in the zone of body cooling), the skin temperature falls to a lower point for the cold-air warm-wall situation, thus decreasing the actual rate of heat loss. The reason for this effect is somewhat obscure, but it is believed to be related to local stimulation by the colder air exerted on the membranes of the nose and throat and to the greater chilling of the exposed skin surfaces when those parts of the body are moved. Studies made at the John B. Pierce Laboratory of Hygiene in this connection indicate surprising influences on decrease in skin temperature exerted by brief cold stimulation of small regions of the body, as by an ice bag on the neck. This is of some practical interest as indicating that warm walls with cool air tend to promote vaso-motor adjustment of the body to the thermal environment, and that extreme local chilling, as by localized drafts, may well be harmful.

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[1] A two-piece suit of cotton underwear, a cotton shirt without tie, socks, low leather shoes, and a dark gray, single-ply suit with three-quarter lined coat and fully lined vest.

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An important point which has been brought out in these studies is the very minor effect of variations in relative humidity in the zone of body cooling. Below an air temperature of 77 F (25 C)—for the clothed, semi-reclining body—sweat secretion is minimal (only a quarter of the total heat loss) and variations downward in the relative humidity of the atmosphere exert only a minor effect upon evaporative heat loss. The reason for this is that the vapor-pressure at the skin surface (at 91 F, or 33 C) is 1.47 in. of mercury; while in air at 70 F (21 C), 80 per cent saturated, it is only 0.58—a gradient which permits full evaporative heat loss. Under conditions of low air movement, and with a relative humidity of 30-35 per cent, the evaporative heat loss is about 4 kilogram-calories per square meter (1.5 Btu per square foot) per hour greater than with a relative humidity of 75-80 per cent. This is the equivalent of increasing the air temperature by less than 1 F.

Excessive atmospheric humidity at low temperatures may exert a chilling effect through increased conductivity of the clothing. General empirical impressions indicate that this factor may be an important one, but it has not as yet been sufficiently studied under laboratory conditions.

Heat losses from the body in the cold zone are, then, influenced chiefly by air and wall temperatures, and very markedly by air movement. The effect of the quantitative influence of air movement has not as yet been fully worked out, and long-time experiments with this end in view are now in progress. It is known, however, that air movement greatly increases the heat loss in a cold environment. Recent studies at the John B. Pierce Laboratory of Hygiene indicate that at an air temperature of 70-72 F (21-22.5 C) an increase in air movement from 17 to 100 linear feet per minute is approximately equivalent to a drop of 7 deg in operative temperature. It is generally assumed, though without exact physiological observations, that moderate and variable air movement is favorable to comfort. An air temperature of 72 F (22.5 C) is definitely comfortable with an air movement of 17 fpm; definitely uncomfortable with movement of 100 fpm. It has generally been found that 50 fpm is an upper limit of comfort for persons at rest.

Results for *storage*, as here cited, refer to the decreasing temperature of body tissues over a two-hour period. It is obvious that this process cannot continue indefinitely, and members of the Committee are now conducting long-time experiments to determine how and when a state of thermal balance between the body and its environment may ultimately be reached.

REACTIONS IN THE HOT ZONE

Passing to the hot zone (above 86 F-30 C for the clothed, semi-reclining body) an entirely different relationship exists. Until the upper limit of adjustment is reached, sweat secretion—through a very perfect physiological adaptation—increases steadily to produce the evaporative heat loss necessary to maintain heat balance between the body and its environment. This process, although successful as a regulatory factor, may be accompanied by symptoms of marked discomfort.

It is important to note that this automatic regulation of thermal interchange in the hot zone is accomplished with remarkable success, however the theoretical heat loss demand may be modified by varying conditions of relative

humidity and air movement as well as of operative temperature. It has, however, a very definite upper limit at the point at which the sweat secretion accumulates and runs off the body without exerting further cooling effect. Studies at the John B. Pierce Laboratory of Hygiene have established upper limits for the clothed, semi-reclining body at an air movement of 17 linear feet per minute as given in Table 1.

These figures should be of interest for comparison with the very valuable studies being made by another Society Research Committee on Air Conditioning in Industry with regard to the upper limits of temperature and humidity which are bearable in the course of various kinds of industrial work. It should be noted that the data cited apply to a resting position with a low metabolic rate, and the quantitative limits of the various zones must certainly vary directly with changes in metabolism.

It should be understood that such figures as those cited may not necessarily represent the limit of human endurance. When a point is reached at which

TABLE 1—UPPER LIMITS OF EVAPORATIVE REGULATION

ENVIRONMENTAL TEMPERATURE		RELATIVE HUMIDITY PER CENT
F	C	
126.9	52.7	0
122.0	50.0	5
113.0	45.0	18
108.5	42.5	26
104.5	40.0	38
99.5	37.5	51
95.0	35.0	69
90.5	32.5	89
87.8	31.0	100

evaporation of sweat can no longer dissipate the heat produced by metabolism, adjustment may be reached (particularly in the acclimatized subject) by an increase in skin temperature and by negative storage. A. Missenard, in observations on nude workers in a potash mine in France, has found that they were able to perform light work for six consecutive hours with an air temperature of 110.5 F (43.6 C) and a wall temperature of 114.5 F (45.8 C) and 40 per cent relative humidity. At high air temperatures, the effect of an air movement of 100 fpm is to decrease discomfort (from its effect on evaporation), even when the air temperature is actually above the temperature of the skin.

In practice, one of the most important problems of air conditioning is stratification of temperature. It is probably important for comfort as well as for heat economy to make the differential between floor and ceiling as small as possible.

Just as the effect of cold air may be balanced by that from hot walls, so Dr. C. A. Mills has pointed out that hot air may be made comfortable by the counter-effect of radiation to cold walls. The quantitative details have not as yet been worked out but are at present under study by committee members.

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It should be pointed out, however, that there is no reason to expect any particular physiological values to be introduced by a situation of this kind, while a combination of hot air and artificially cooled walls would be one which would be highly costly to maintain in practice.

CALORIMETRIC EXPERIMENTS ON NUDE SUBJECTS

The results reported, from experiments with clothed subjects at the John B. Pierce Laboratory of Hygiene at New Haven, are in substantial accord with a series of researches conducted at the Russell Sage Institute of Pathology in New York. These observations have been carefully made in a calorimeter which records simultaneously the heat production, the heat losses, the vaporization, radiation, and convection. The subjects' notes of mental state are also preserved. Studies have been made on effects of fall in body temperature and on the normal chill mechanism; and on the effects of alcohol and of hot and cold drinks; and on the sensory thresholds to hot and cold stimuli.

In all work on air conditioning it is of prime importance to make proper allowance for the clothing and activity of the subjects. This is brought out in Fig. 2. The first three diagrams are based on studies in the respiration calorimeter showing the physical and physiological changes with rising temperature. The fourth is based on only two experiments. The fifth and sixth are largely extrapolations, as the apparatus was not suitable for measurement of metabolism more than three times the basal level.

It is suggested that diagrams of this type may serve as a basis for a study of the physiology of air conditioning. For example, an analysis of the diagrams shows the following.

I. Reactions of Body to Different Environmental Temperatures. For Nude Subjects Lying in Bed.

1. Cold Zone 83 F (28 C) and below.
   No control of heat loss; body temperature maintained by chills.
   Blood vessels of skin are maximally constricted throughout the zone; body cools as a physical object in accord with Newton's Law.
   Evaporative loss = one-half respiratory and one-half seepage through skin.
   = constant.
   Chills brought on periodically when mean body temperature has fallen 1.2 F (0.65 C). Alcohol anesthetizes chill mechanism. Chill with alcohol occurs after a degree or more fall in temperature.

2. Zone of vaso-motor control 83 F-88 F (28 C-31 C) (Comfort Zone).
   Subjects most comfortable; no progressive changes in body temperature. Blood flow to the skin changes so that heat loss is automatically adjusted to the environment. No generalized feelings of warmth or cold; no sweating. Chills are easily evoked with cold drinks. Sweating is evoked by hot drinks.

3. Zone of vaso-motor and evaporative regulation 88 F-98 F (31 C-36 C).
   Peripheral blood flow increases in proportion to temperature. Sweating (at constant low humidity), in proportion to temperature. Body cooling is adequate; general level of body temperature is higher, pulse rate increases in proportion to temperature. Body is continually stimulated by sensations of heat which are uncomfortable, but which are necessary to thermal regulation. At higher temperatures, even though the body may be able to cool itself from a physical standpoint, reactions to heat may be anomalous, causing heat stroke and exhaustion.

II. It is important to note that a rise in skin temperature of 0.0025 C will evoke a sensation of warmth in three seconds, and a fall in skin temperature of 0.004 C in one second will produce a cold sensation. The peripheral sense of heat or cold is not changed by exposing more of the body if the head is once exposed. The head (face and neck) are the most sensitive portions of the body to thermal stimuli. The backs of the hands are next. Calorie for calorie, *cold* radiation causes twice the skin temperature change that is produced by *heat* radiation. Local vaso-motor response to thermal stimuli occurs after two minutes at points stimulated. The result is that cold radiation causes a three- to four-fold greater change in exposed skin temperature than does heat radiation.

The threshold for pain is lower for cold than for heat and of a different quality. Various types of heat radiation have different sensory thresholds.

The following differences are noted between men and women. In the cold zone the thickness of the thermal insulating tissues of women is almost double that of men, although the sensory responses to cold are similar. In the hot zone the threshold to sweating is higher and skin temperature levels are also higher.

PROBLEMS OF COMFORT

It will be noted that the operative temperatures cited for thermal equilibrium appear distinctly high as compared with the commonly accepted standard of 68-70 F (20-21 C) for schools and similar occupied spaces. It should be expected that in the long run the comfort zone must coincide with the area of thermal adjustment with minimum physiological effort, and this adjustment, as noted, is reached for the clothed subject in a semi-reclining position, at approximately 80 F (26.7 C) operative temperature, corresponding to 74 F (23.3 C) effective temperature (with 50 per cent relative humidity).

In the very valuable field studies made by the Society at Pittsburgh, Minneapolis, Toronto, and Texas A. & M. College in regard to summer cooling requirements, effective temperatures of 71 F (21.7 C) or thereabouts, corresponding (with 50 per cent relative humidity) to an operative temperature of 76 F (24.4 C), seem to be preferred.

On analysis these results do not seem inconsistent with each other, or with earlier practice. Both physiological adjustment and sensations of comfort will vary with three factors; the heat production of the organism (metabolism), the clothing worn, and the acclimatization associated with the season of the year.[2] Such results as those obtained at the Pierce Laboratory, which indicate an optimum around 80 F (26.7 C) operative temperature, will obtain in a semi-reclining position, with metabolism at a low level and with light clothing. The fact that the ASHVE experiments on summer cooling were made with subjects having a substantially higher metabolism (since they were at work and not reclining), should account for the 4 F (2 C) lower optimum observed. Earlier observations in school rooms pointing to a desirable operative temperature of 70 F (21.1 C) or below were obtained with a physical activity more or less comparable with that of less active factories, with relatively high metabolism on account of the fact that the subjects were children, and with heavy

[2] Reference is here, of course, made not to basal metabolism (minimal) which is relatively constant, but to the actual mean daily metabolism as affected by the activity of the subject.

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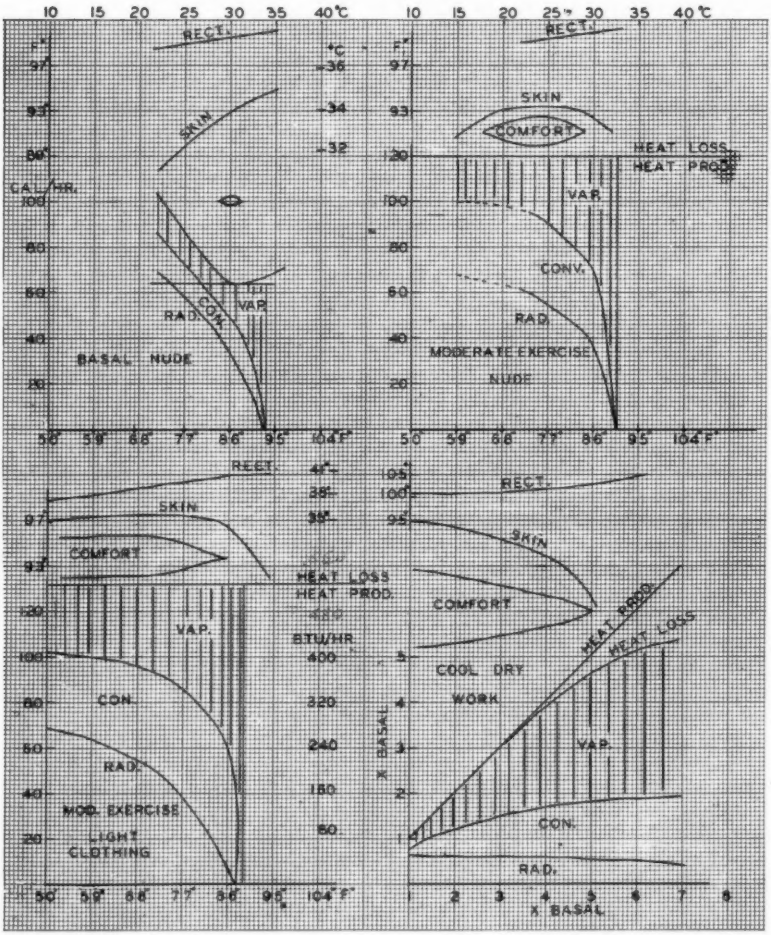
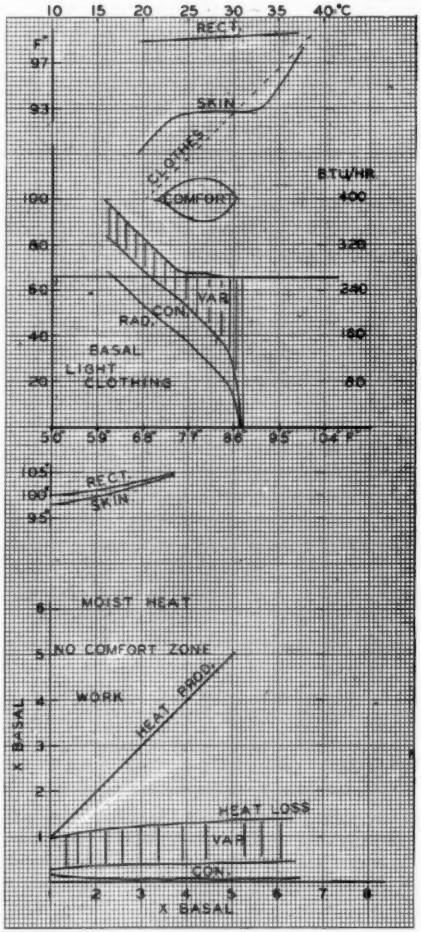


FIG. 2. THE RELATION OF ATMOSPHERIC TEMPERATURE AND METABOLISM TO VARIOUS NUDE



TYPES OF HEAT LOSS AND COMFORT FOR THE SUBJECT

In the first four diagrams, the ordinates represent rectal and skin temperatures (*above*) and calories of heat production and heat loss (*below*); the abscissae represent air and wall temperatures, which were identical. Heat loss is divided into radiation, convection and vaporization. In the last two diagrams, the ordinates represent heat production and heat loss using the basal (minimal) metabolism as a unit equal to one; the abscissae represent activity of the subject in terms of basal metabolism. Twice the basal value corresponds to walking slowly and six times the basal value to heavy labor.

clothing, and acclimatization to winter conditions. (All these data are associated with low air movement.)

It is interesting to note that the New York State Commission on Ventilation found operative temperatures below 55 F (12.8 C) preferred by students in a gymnasium. It seems of fundamental importance that the comfort zone may vary over a range of 25 F (14 C), depending on the physical activity, the clothing and the acclimatization of the subject.

METHODS OF AIR EXAMINATION

In connection with studies of atmospheric conditions from the viewpoint of human health and comfort, attention should be called to the following recommendations of the Sub-Commission on the Hygiene of Environmental Conditions in the Dwelling, of the Housing Commission of the League of Nations, adopted at Geneva on June 25, 1937:

"Valuable as these various instruments[a] may be in establishing the conditions of comfort in any given enclosed space, it is of great importance in all research studies to make an accurate record of each of the four independent factors governing bodily heat exchange, temperature, movement and humidity of the air, and mean radiant temperature of the surrounding surfaces. For this purpose we suggest in the interest of comparability the use of the following four types of instruments or others yielding similar data:

(*a*) Silvered dry-bulb thermometers or hair-pin thermometers (Bargeboer).
(*b*) Silvered dry Kata thermometers or the hot wire anemometer.
(*c*) Psychrometer, wet- and dry-bulb, whirling or ventilated.
(*d*) Globe thermometer (Vernon) or the dry resultant thermometer (Missenard).

Such instruments as these, when properly calibrated and their readings are compared, can be used for determining the four basic physical factors concerned separately or in certain combinations. The results of the four physical measurements thus determined can generally be translated into the terms of any special instrument combining two or more of them.

The single indices of thermal conditions mentioned above may be found to be of great utility for discussions in the countries of their adoption. It would, however, facilitate comparisons with the scales of measurement used in other countries if the measurements of the separate thermal factors were published along with those of the single index.

In some instances it may be important to record not only the movement and temperature of the air at various levels, but also the temperature of each wall and window, of the flooring, and of the ceiling, and to measure the total effective radiation of the surroundings in six directions—in order to trace the exact causes of defects in the building which have an unfavorable influence on the heat exchanges of its inhabitants. Facts of this type are of great practical importance."

The ASHVE Technical Advisory Committee on Physiological Reactions endorses these viewpoints and urges that in all studies of air conditions records be obtained showing:

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[a] Refers to various instruments (such as the eupatheoscope and the thermo-integrator) designed to measure the combined influence of two or more of the factors of air temperature, wall temperature, relative humidity, and air movement.

(*a*) True air temperature (free from radiation effects).
(*b*) Air movement.
(*c*) Humidity.
(*d*) Mean radiant temperature of surrounding surfaces.

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No. 1108

CARDIAC OUTPUT, PERIPHERAL BLOOD FLOW, AND BLOOD VOLUME CHANGES IN NORMAL INDIVIDUALS SUBJECTED TO VARYING ENVIRONMENTAL TEMPERATURES

By FORD K. HICK, M.D., PH.D.,* ROBERT W. KEETON, M.D.,** NATHANIEL GLICKMAN, M.S.,*** AND HARRY C. WALL, M.D.,† CHICAGO, ILL.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the Department of Medicine of the University of Illinois.

THE human body maintains a nearly constant temperature by balancing its heat production against heat loss. DuBois has pointed out that in a man [1] millions of calories of heat produced are exactly balanced by heat loss over a period of years with surprising facility. In the *comfort zone* [2] a human maintains normal temperature and a warm skin while losing 25 per cent of his heat by evaporation.[3] In the *zone of cooling* [4] a man loses heat largely through radiation until he chills, begins to shake, and raises his heat production to recover his normal temperature. In the *zone of evaporative regulation*,[5] he loses more than 25 per cent of his heat through the evaporation of water, and maintains a normal body temperature until this mechanism fails. The whole study of heat regulation is then one of the physiological adjustments of the body to meet a physical problem in thermodynamics. When these adjustments occur smoothly and escape the attention of the individual he is *comfortable*. When they proceed with more difficulty he is *uncomfortable*.

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* Department of Medicine, University of Illinois.
** Head, Department of Medicine, University of Illinois.
*** ASHVE Research Fellow, Department of Medicine, University of Illinois.
† Department of Medicine, University of Illinois.

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Presented at the 45th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Pittsburgh, Pa., January, 1939.

Comfort then becomes an index of the easy adaptation of the heat loss mechanism to the needs of the moment.

In studies by DuBois, Winslow, and associates, many facts relative to heat loss through the skin have been detailed. They have noted that the delivery of heat from the deeper parts of the body to the surface is accomplished chiefly by the circulation of the blood. It, therefore, seemed imperative to make a quantitative study of the circulatory adjustments concerned in the transport and dissipation of heat. The present report details observations on the measurements of cardiac output, the quantity of blood flowing through the periphery, and the circulating blood volume. It is obvious that man must be used as the experimental subject in the physiology of heat loss, since he differs so widely from animals.

PLAN OF EXPERIMENT

Healthy subjects were studied under basal conditions in an atmosphere within the range of the *comfort zone*. When subjects have fasted for 12 hours and have remained quiet for one to two hours after a good night's rest they are under basal conditions. These subjects have a constant heat production varying little from one day to another.

The atmosphere was next changed to the *zone of evaporative regulation* so that all the heat was lost by evaporation. The humidity governs the ease of heat loss in this evaporative zone, so the subjects were submitted to hot dry and hot wet conditions. All measurements were carried out two to three times on adult, normal subjects, one male and one female.

The general procedure was such that the subject was in the hospital overnight, rested in the air conditioned room if it was comfortable, and remained quiet in the early morning until the work for that day was completed. If the room was uncomfortable, the subject slept in another room in the hospital and was wheeled into the air conditioned room between 5 and 6 a.m. and observations began two hours or more afterward. All observations were therefore under post-absorptive or *basal* conditions and with a nude male subject and a practically nude female subject. The subject was on a bed.

PROCEDURE

The man's age was 22, height 74¾ in., weight 168 lb, DuBois surface area 2.0 square meters. The woman's age was 24, height 63 in., weight 105 lb, DuBois surface area 1.45 square meters.

Observations were made at three general conditions: a comfortable condition (about 84 F dry-bulb and 60 F wet-bulb), a hot dry, and a hot wet condition. To insure that all the heat loss would be by evaporation in the hot conditions, the dry-bulb was set at 99.5 F and the wet-bulb adjusted to between 65 to 74 F for dry, and 85 to 91 F for wet conditions. The two hot periods differ only in moisture content. Air movement was kept constant and minimal. No drafts which would activate a precision balanced vane type anemometer were detected. The temperature of the walls was uniformly equivalent to air dry-bulb temperature.

Cardiac output was measured by the method of Grollman[6] as modified by McMichael[7] to use the Van Slyke gas analytic apparatus. The oxygen consumption of the subjects was determined by analysis of expired air collected in a spirometer (Tissot tank). Skin temperature was obtained by a multiple lead constantan-copper thermocouple apparatus patterned directly after that of Sheard,[8] using a potentiometer for measurement of electro-motive force (E. M. F.). The mean skin temperature, an average figure weighted according to the surface area of the parts studied, was calculated in the manner of Hardy and DuBois,[9] after measurement of 16 separate points. Blood volume studies were carried out using a technique described by Gibson and Evelyn[10] with surprising ease in reproducing results.

In determining venous oxygen saturation blood was drawn from a superficial vein at the elbow without a tourniquet. The samples were collected without contact with air and analyses were carried out as described by Van Slyke et al.[11]

Measurement of the rate of insensible loss of weight was carried out on a balance sensitive to less than one gram.

RESULTS

Cardiac Output

Table 1 lists the essential data on cardiac output on these subjects. The rectal temperatures show anticipated variations. Subjectively the hot wet atmosphere was extremely uncomfortable. In this condition the elimination of heat is difficult and at times there is an actual retention. This is shown by the elevation of the rectal temperature above 98.6 F. With a retention of heat there is an elevation of the oxygen consumption and a corresponding increase in cardiac output. Unless such a retention of heat occurred there was no change in the cardiac output. The pulse rate is very low under comfortable conditions on these two subjects. It is somewhat higher in the hot dry conditions in the woman. Obviously the pulse rate changes do not parallel changes in cardiac output.

Blood Flow Through the Periphery

A. Conductance: Tissues without circulating blood have a low or negligible power of thermal conduction. In fact, experimental work[12] indicates that the

[6] The Cardiac Output of Man in Health and Disease, by Arthur Grollman. (Thomas, Springfield, Ill., 1932.)

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[8] The Electromotive Thermometer: An Instrument and a Method for Measuring Intravenous, Superficial, and Cavity Temperatures, by Charles Sheard. (*American Journal of Clinical Pathology*, 1:209, 1931.)

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[10] Clinical Studies of the Blood Volume IV. Adaptation of the Method to the Photoelectric Microcolorimeter, by J. G. Gibson, II, and K. A. Evelyn. (*Journal of Clinical Investigation*, 17, 153, 1938.)

[11] Quantitative Clinical Chemistry, by J. P. Peters and D. D. Van Slyke. (Vol. II, p. 337, Williams and Wilkins Co., Baltimore, 1932.)

[12] The Physical Laws of Heat Loss from the Human Body, by J. D. Hardy. (*Proceedings* of the *National Academy of Sciences*, 23, 631, 1937.)

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TABLE 1—CARDIAC OUTPUT

SUBJECT	CONDITION	DRY-BULB, TEMPERATURE, F	WET-BULB, TEMPERATURE, F	OXYGEN CONSUMPTION, CC/MIN	ARTERIO-VENOUS OXYGEN DIFFERENCE	CARDIAC OUTPUT, L/MIN	RECTAL TEMPERATURE, F	PULSE
J. Morgan (Female)	Comfortable	84.0	63.5	188.5–187.5	69.5	2.7	...	64
		84.0	64.6	185.0–178.5	77.7	2.4	98.4	63
		84.0	61.7	178.3–190.5	64.9–76.5	2.7–2.5	97.9	63
		Average		184.7	72.1	2.6	98.2	63
	Hot and Dry	99.4	65.0	185.5–188.3	61.3–73.4	3.0–2.6	98.8	70
		99.4	67.5	181.3–183.0	66.4–73.5	2.7–2.5	98.4	74
		99.5	71.6	185.8–182.2	62.0	2.9	98.4	68
		Average		184.4	67.3	2.7	98.5	71
	Hot and Wet (No fever)	98.7	84.5	176.7	66.0	2.7	98.8	74
	Hot and Wet (Fever)	99.0	87.2	185.0	52.0–57.1	3.5–3.2	99.2	74
		98.8	86.1	193.3	52.8	3.7	99.2	75
		99.5	90.0	215.3	51.1–55.9	4.2–3.9	99.5	87
		Average		197.9	53.8	3.7	99.3	79
T. Schertz (Male)	Comfortable	87.0	65.8	286	70.5–65.0	4.1–4.4	97.9	55
		85.8	63.8	268–252	60.3–65.3	4.3–4.0	97.4	48
		Average		268.6	65.2	4.2	97.7	52
	Hot and Dry	101.0	75.5	261.5–280	56.0–55.5	4.8–4.9	99.1	49
		99.5	64.5	254.0–245.3	54.3	4.7	98.5	47
		Average		260.2	55.3	4.8	98.8	48
	Hot and Wet (Fever)	100.0	83.0	285.0–287.0	67.1–68.3	4.6–4.2	99.0	56
		99.3	86.0	248.2–266.2	39.4	6.3–6.8	99.4	68
		101.0	90.0	345	50.7–55.5	6.8–6.2	101.4	80

body tissue is a good insulator. Virtually all the heat reaching the body surface is brought by the blood and so the heat flux becomes an index of blood flow through the surfaces. The quantity of heat which must be eliminated depends on the metabolism of the tissues, which, of course, is determined by the oxygen consumption. *Conductance of the Superficial Tissues* is a concept that has been used by both Hardy and DuBois [13] and Winslow et al;[14] and has been formulated to include the factors mentioned. The formula of Winslow is as follows:

$$c = \frac{M \pm S}{A(T_B - T_S)}$$

where M is heat produced and S is heat stored, both expressed in kilogram calories per unit time. A is the area of the body as designated by the DuBois formula.[15] T_B is body or rectal temperature (Centigrade). T_S is mean weighted skin surface temperature (Centigrade). In Table 2 the *conductance values* for the two subjects are shown. In computing these values the following formula has been used:

$$C = \frac{M}{T_R - T_S}$$

where M is total metabolism of the subject expressed in kilogram calories per minute, T_R is rectal temperature, and T_S is weighted mean body surface temperature in Centigrade.

The heat loss in the conductance formula was taken to be equivalent to the heat production as measured by oxygen consumption. This is, of course, an approximation, which is justified by the absence of change in rectal temperatures. Storage therefore, may be neglected. The conductance is figured for the subject as a whole rather than for a unit area. Hence the factor A is omitted. Conductance values derived by this formula for an individual cannot be compared with values from other individuals.

It is apparent at a glance that the conductance or the peripheral blood flow rises as the mean skin temperature rises. This rise from the comfortable to the hot dry atmosphere is of considerable magnitude, and would indicate that the peripheral blood flow was increased 250 per cent. The increase in conductance from a hot dry to a hot wet atmosphere is not so great (35 per cent) in subject Morgan and even less in subject Schertz.

B. Venous Oxygen Saturation: Table 3 presents venous oxygen saturation results on five healthy male subjects. This measurement has been used in the

[13] Loc. Cit. See Note 4.
[14] Physiological Reactions of the Human Body to Varying Environmental Temperatures, by C.-E. A. Winslow, L. P. Herrington and A. P. Gagge. (*American Journal of Physiology*, 120:1, 1937.)
[15] A Formula to Estimate Approximate Surface Area If Height and Weight Be Known, by E. DuBois and E. F. DuBois. (*Archives of Internal Medicine*, 17:863, 1916.)

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TABLE 2—PERIPHERAL BLOOD FLOW—CONDUCTANCE

SUBJECT	CONDITION	DRY-BULB TEMPERATURE, F	WET-BULB TEMPERATURE, F	OXYGEN CONSUMPTION CC/MIN	MEAN SKIN TEMPERATURE, F	RECTAL TEMPERATURE, F	$T_R - T_S$	CONDUCTANCE KILOGRAM CALORIES/ DEG C/MIN	H_2O LOSS, GRAMS/MIN
J. Morgan (Female)	Comfortable	84.5	64.5	...	93.7	98.4	4.7	...	0.442
		84.0	64.6	185.0–178.5	92.4	97.5	5.1	0.30	0.438
		83.4	62.0	...	92.3	98.2	5.9	...	0.511
		84.0	63.5	188.5–187.5	92.8	...	...	...	0.442
		84.0	61.7	178.3–190.5	92.7	97.9	5.2	0.30	0.460
		Average		184.7	92.8	98.0	5.3	0.30	0.459
	Hot and Dry	99.4	65.0	185.5–188.3	96.8	98.8	2.0	0.82	2.234
		99.4	67.5	181.3–183.0	96.3	98.4	2.1	0.73	2.235
		99.8	71.9	...	96.7	98.9	2.2	...	2.584
		99.5	71.6	185.8–182.2	96.2	98.4	2.2	0.74	2.457
		Average		184.4	96.5	98.6	2.1	0.76	2.378
	Hot and Wet (No fever)	98.7	84.5	176.7	97.3	98.8	1.5	1.07	2.777
		99.5	81.5	...	96.9	98.8	1.9	...	2.362
		99.5	81.0	...	97.1	98.6	1.5	...	2.236
		Average		176.7	97.1	98.7	1.6	1.07	2.458
	Hot and Wet (Fever)	98.8	86.1	193.3	97.8	99.2	1.4	1.33	2.313
		99.0	87.2	176.0	97.6	99.2	1.6	0.94	2.841
		99.5	90.0	215.3	98.0	99.5	1.5	1.30	3.774
		Average		194.9	97.8	99.3	1.5	1.19	2.976
T. Schertz (Male)	Comfortable	85.8	63.3	268.0–252.0	92.9	97.4	4.5	0.50	1.36
		86.0	63.5	272.0	93.8	98.2	4.4	0.52	1.33
		87.0	65.8	286.0	93.8	97.9	4.1	0.60	1.28
		Average		269.0	93.5	97.8	4.3	0.54	1.32
	Hot and Dry	99.5	64.5	254.0–245.3	96.6	98.5	1.9	1.10	3.30
		99.5	68.1	...	96.4	98.2	1.8	...	...
		99.5	70.0	251.0	96.9	98.6	1.7	1.35	3.46
		99.0	73.0	...	96.8	98.5	1.7	...	...
		99.5	75.0	...	96.9	98.5	1.6	...	...
		Average		250.1	96.7	98.5	1.7	1.22	3.38
	Hot and Wet	100.0	84.0	285.0–282.0	97.3	99.2	1.9	1.37	5.81
		98.7	84.8	...	97.1	99.3	2.2	...	5.20
		98.8	85.0	...	97.3	99.4	2.1	...	5.50
		99.3	86.0	266.2	97.9	99.4	1.5	1.61	...
		Average		277.7	97.4	99.3	1.9	1.49	5.47

past to study rate of blood flow through an extremity in man.[16] Since oxygen consumption, or heat produced in an extremity, does not change under different environments, the oxygen saturation figures on venous blood become an index of rate of blood flow. Arterial blood normally is 95 per cent saturated with oxygen. Under the comfortable conditions the venous blood bears about 70 per cent saturation with considerable individual variation. This was observed to be elevated significantly in four of the five subjects under hot dry conditions. Under hot wet conditions the saturation of venous blood approximates arterial

TABLE 3—PERIPHERAL BLOOD FLOW—VENOUS OXYGEN SATURATION

SUBJECT	CONDITION	DRY-BULB, TEMPERATURE, F	WET-BULB, TEMPERATURE, F	VENOUS OXYGEN SATURATION, PER CENT	HEMOGLOBIN, GRAMS PER 100 CC
G. Brebis	Comfortable	84.8	56.5	54.8	16
	Hot Dry	99.5	62.7	89.9	14.6
	Hot Wet	98.8	90.8	91.0	14.4
Gregoric	Comfortable	84.7	56.5	74.5	13.2
	Hot Dry	99.5	62.7	91.4	13.3
	Hot Wet	98.8	90.8	93.4	14.2
McKeever	Comfortable	84.7	56.5	65.8	13.5
	Hot Dry	99.5	62.7	81.2	14.1
	Hot Wet	98.8	90.8	93.0	14.8
Carlson	Comfortable	82.5	60.0	70.4	15.5
	Hot Dry	99.2	62.2	73.2	16.6
	Hot Wet	99.5	90.3	84.0	16.5
Schertz	Comfortable	81.5	62.0	66.0	13.8
	Hot Dry	99.5	64.5	77.2	...
	Hot Wet	...	...	81.0	...

blood in three of the five subjects. Apparently the circulation through an extremity can exceed in extreme heat eight times that under comfortable conditions.

These data supplement the data on conductance in that the hands (probably the feet) can open capillary beds widely and tolerate a maximal rate of blood flow through them. The conductance values mentioned do not show such a great change in blood flow, because they are derived from a mean skin temperature weighted to include the surfaces of the body which cannot tolerate excessive flow of blood through the skin. In general the skin over the trunk, thighs, and shoulders does not have the number of available capillaries to open during hot exposure which are found in the skin of the hands.

Water loss is shown in Table 2 in grams per minute. In the hot dry condition where the sweat produced approaches the amount lost in the hot

[16] Intermittent Venous Hyperemia in the Treatment of Peripheral Vascular Disease, by G. DeTakats, F. K. Hick and J. C. Coulter. (*Journal American Medical Association*, 108:1951, 1937.)

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wet condition, the skin of the female subject was usually dry to the touch and definitely cool. The male subject produced excessive quantities of sweat in the hot wet environment, much of which ran off without evaporating. He, too, usually had most of his body dry to the touch while in the hot dry condition. Obviously the quantity of water lost through sweating bears no constant relationship to the rate of blood flow through the skin as measured by conductance or venous oxygen saturation. There is some relationship of heat loss by evaporation and circulation through the skin until free sweating appears.

C. Blood Volume: Blood volume studies have been completed on a male and female subject under hot wet and hot dry, as well as comfortable conditions. The results are presented in Table 4. There is an observed rise of about 8 to 12 per cent in these two subjects in hot wet conditions. Bazett [17]

TABLE 4—BLOOD VOLUME STUDIES

SUBJECT	CONDITION	DRY-BULB, TEMPERATURE, F	WET-BULB, TEMPERATURE, F	PLASMA VOLUME, CC	CIRCULATING BLOOD VOLUME, CC	HEMATOCRIT	PULSE
J. Morgan	Comfortable	83.4	62.0	2342	3747	37.5	63
	Hot and Dry	99.8	71.9	2453	3882	36.8	70
	Hot and Wet	98.7	84.5	2778	4375	36.5	74
T. Schertz	Comfortable	81.5	62.0	4651	8189	43.2	48
	Hot and Dry	99.5	68.1	4808	8108	40.7	50
	Hot and Dry	101.0	75.5	4902	8239	40.5	49
	Hot and Wet	100.0	83.0	5263	8831	40.4	56

has observed a rise in circulating blood volume in hot conditions after days of exposure. The results presented in this paper are after only a few hours, two or three, of hot wet conditions. Apparently the body calls on its store of water to raise the blood volume as the capillary bed in the skin dilates. This was independent of fever for the rectal temperature was 99 F in Schertz and 98.8 F in Morgan on the days of the hot wet conditions.

DISCUSSION OF RESULTS

It appears that the rate of blood flow through the skin governs its temperature and that the skin temperature is the delicate mechanism for controlling loss of heat to meet the needs of the individual. The skin being dry or wet, however, depends on the humidity in the presence of a high dry-bulb temperature. With a low humidity in the evaporative zone the skin temperature is lower than with a high humidity, the dry-bulb remaining the same. This has been long well known and accounts for the ability of men to work in deserts, and the difficulty of hard physical work in the tropics.[18]

[17] Cardiovascular Changes in a Hot Room, by H. C. Bazett, J. C. Scott and F. W. Sunderman. (*American Journal of Physiology*, 123:11, 1938.)
[18] Life, Heat and Altitude, by D. B. Dill. (Harvard University Press, Cambridge, 1938.)

That the cardiac output does not rise as soon as the pulse rate is a matter of considerable interest. It would appear that the body *should* possess a mechanism of increasing the cardiac output to facilitate greater movement of blood through the skin in the presence of great heat. The data do not show this, however. These observations are quite limited in number but appear significant.

Christensen [19] found that in work cardiac output is a function of the heat production and does not rise until heat production rises. Data resulting from the studies reported here support the concept that with rising environmental temperature the skin temperature rises as a result of deflection of blood through the skin. This does not entail an increased cardiac output and may not require an increased pulse rate. In fact the pulse rate may rise without the cardiac output rising, which constitutes a type of inefficient heart action.

As the rectal temperature rises the heat production rises and cardiac output goes up.

The requirement of heat loss does not dictate cardiac output. It does dictate mean skin temperature and rate of blood flow through the skin.

CONCLUSIONS

The cardiac output does not rise on exposure of a resting man to hot conditions until his metabolism (heat production) rises.

The blood flow through the periphery, particularly the hands, becomes very great as the mean skin temperature approaches the rectal temperature.

The blood volume rises within periods of two or three hours exposure to hot wet conditions.

Cardiac output is governed by heat production.

The skin temperature can be duplicated from one day to another in controlled rooms. It is the delicate mechanism governing heat loss. In the evaporative zone it is partly governed by humidity.

Venous oxygen saturation in superficial veins is an index of rate of blood flow through an extremity. Figures as high as 93 per cent of saturation were observed in hot wet conditions.

ACKNOWLEDGMENT

Acknowledgment is made of the cooperation of the Engineering Experiment Station of the University of Illinois, Urbana, Ill., through engineering consultation and services rendered by Prof. A. E. Hershey and Prof. M. K. Fahnestock of the Department of Mechanical Engineering.

[19] Arbeitsphysiologic, by E. H. Christensen. (5:479, 1932.)

DISCUSSION

DR. E. F. DUBOIS[20] (WRITTEN): These studies of Dr. Hick and his associates supply data of great value for physiologists and air conditioning engineers. There are a fairly large number of experiments in the literature available now giving the external factors of heat loss and conductance through the skin and subcutaneous tissues but very few showing cardiac output and blood volume with different environmental conditions.

The skin temperatures found by these investigators in the comfort zone correspond closely with ours measured by another method. Their figures for conductance in the comfort zone are about 60 per cent higher than ours. Some of this discrepancy may be due to the higher humidity in their experiments, some to the higher metabolism of their subjects. In our calculations we almost always have to make allowance for the storage or loss of heat in the body, especially in the region near the skin.

We would like to point out the fact that rise in skin temperature does not necessarily mean rise in peripheral blood flow. For example in exercise the skin temperature may fall as a result of sweating at a time when the peripheral blood flow has increased markedly. At rest with a cold environment skin temperature may fall at a time when there is little if any change in peripheral flow. Also at rest in the warm zone skin temperature may change little at a time when peripheral blood flow is rising rapidly.

W. L. FLEISHER: I did not intend to enter into this discussion but as it relates to the work we have done in the Laboratory I feel that in a way it is incomplete. In Table 2, under hot and wet, we have in most cases a wet-bulb which is relatively lower than any wet-bulb of which we count any particular change. I think that it probably would be well to carry this experiment further, so that the condition stated as hot and dry be related to the same wet-bulb as those experiments specified as hot and wet. The changes taking place seem to correspond very definitely with the work that we discovered in our own Laboratory on our experiments in air conditioning in industry. The changes do not take place particularly until the conditions rise above 85 or 86 wet-bulb, which is a condition, I think, that Haldane discovered in about 1903.

All of the indications are that the conditions occurring as physiological changes taking place occur practically as we have discovered and as other investigators have discovered, over a period of about 35 or 40 years. Therefore, I think that instead of designating as hot and dry and hot and wet, we should relate them to the same wet-bulb or the same effective temperature conditions. In Laboratory results previously given, I think we indicated pretty clearly that no physiological changes took place or that physiological changes rather did take place at the same effective temperature irrespective of the relative humidity if the effective temperature was the same, and that until we make that investigation to carry that experiment that far we will not have the full facts on which we can predicate results.

PROF. C.-E. A. WINSLOW: I am impressed on the whole with the excellent correlation between the results obtained in these various experiments and it is very important,

[20] Physician-in-Chief, New York Hospital, and Professor of Medicine, Cornell University Medical College, New York, N. Y.

of course, to have that type of corroborative evidence. The work that has been done in Illinois on the conductance of the skin, for instance, fits in very nicely with the results already reported and with the earlier results obtained in the John B. Pierce Laboratory of Hygiene.

There are two things, however, Dr. Keeton has suggested that are new here. One of them is the study that has been made of blood volume, and I would like to call your attention, as an illustration of the way these different researches interlock, to the possible relation of those observations on blood volume to the data presented with regard to increased leucocyte count at high temperatures. If in the experiment at the ASHVE Laboratory there was an increase in blood volume, the leucocyte increases are much more striking than they appeared at first sight and that illustrates very well the way in which these studies interlock.

The other thing that is quite new in this study is the local change in blood volume as measured by venous oxygen saturation. All these conductance indices that we have used in the past have been for the body as a whole, but this method that Dr. Keeton has used makes it possible to show, roughly, at any rate, how these conductance changes vary in different parts of the body.

I think all in all, as Chairman of the Committee on Physiological Relations, that it is appropriate to express again, as I have so often before, the keen appreciation of those of us who approach this subject primarily from a biological standpoint of the wisdom and farsightedness and generosity of the Society in supporting work of this type. It is certain that sound knowledge of these physiological behaviors of the body are essential to progress in heating and ventilation, but nevertheless I think it is quite remarkable that a group primarily of practical engineers should have realized this so fully, and I am sure, as I say, that will prove to be a wise policy.

DR. F. K. HICK: I am grateful indeed for the discussion of this paper. Dr. DuBois, of course, is the pioneer on the quantitation of heat loss studies, having begun that work many years ago in calorimeter experiments, and any note from him in discussion of a paper of this nature is much appreciated.

He makes quite a point in his discussion that rise in skin temperature does not necessarily mean an associated rise in blood flow through that surface. I think that point is very well taken, but open to some controversy. How to tell what changes in skin temperature reflect blood flow changes through that surface and what changes in skin temperature do not reflect such blood flow changes is something I should like to know very much more about. Maybe Dr. Sheard will have something to say on that subject.

With regard to Mr. Fleisher's discussion, we are well aware, this work is incomplete, and we have made no particular efforts to refer it to the effective temperature scale or to particular wet-bulb temperature. We chose the hot and dry situation as one in which the body was losing all of its heat by evaporation at a temperature where that evaporation could be easily accomplished, where the skin of these subjects felt dry to the hand. In the hot and wet condition, of course, they were dripping with sweat. Their skin was wet all over and the perspiration was running off them. It was a situation somewhat like the desert situation in the southwest, for instance, where it is easy to stay comfortable with high dry-bulb if you can evaporate your sweat, and the tropical situation where still air that is very moist permits you to lose relatively little heat by ready evaporation, a distinction that is brought out particularly well in this interesting book by Dill of Harvard entitled, "Life, Heat and Altitude."

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Professor Winslow, in his discussion, mentions the possibility of quantitating blood flow through extremities, a very interesting field, but the exact quantitative interpretation of venous oxygen studies has not conclusively been worked out.

I think Dr. Keeton and I appreciate very deeply the attention that the Society is willing to give to what is after all basic human physiology data.

No. 1109

THE RÔLE OF THE EXTREMITIES IN THE DISSIPATION OF HEAT FROM THE BODY IN VARIOUS ATMOSPHERIC AND PHYSIOLOGICAL CONDITIONS

By CHARLES SHEARD, PH.D.,* MARVIN M. D. WILLIAMS, PH.D.,** GRACE M. ROTH, PH.D.,*** AND BAYARD T. HORTON, M.D.,† ROCHESTER, MINN.

THE human body adjusts itself to low environmental temperatures chiefly by constriction of the peripheral blood vessels and to high atmospheric temperatures largely by an increase of sweat subsequent to maximal vasodilatation of the peripheral vascular system. The investigations of Burton,[1] Burton and Bazett,[2] and Winslow, Herrington and Gagge,[3] demonstrated that the physical regulation, through constriction or dilatation of blood vessels, is due essentially to a change in conductivity over the gradient between environmental and internal body temperatures. That various portions of the body play relatively different rôles in the elimination of heat from the body, and that increasing environmental temperature or increasing heat production causes a much greater shift of blood to the extremities than to other segments of the body, was shown by Maddock and Coller,[4, 5] and Herrick and Sheard.[6] The

* Director, Division of Physics and Biophysical Research, The Mayo Foundation and The Mayo Clinic.
** Associate, Division of Physics and Biophysical Research, The Mayo Foundation.
*** Associate, Section on Clinical Physiology, The Mayo Clinic.
† Consultant, Division of Medicine, The Mayo Clinic.

[1] The Application of the Theory of Heat Flow to the Study of Energy Metabolism, by A. C. Burton. (*Journal of Nutrition*, 1936, *7*:497.)

[2] A Study of the Average Temperature of the Tissues, of the Exchanges of Heat and Vasomotor Responses in Man by Means of a Bath Calorimeter, by A. C. Burton and H. C. Bazett. (*American Journal of Physiology*, 1936, *117*:36.)

[3] Physiological Reactions of the Human Body to Varying Environmental Temperatures, by C.-E. A. Winslow, L. P. Herrington and A. P. Gagge. (*American Journal of Physiology*, 1937, *120*:1.)

[4] The Rôle of the Extremities in the Dissipation of Heat, by W. G. Maddock and F. A. Coller. (*American Journal of Physiology*, 1933, *106*:589).

[5] The Function of Peripheral Vasoconstriction, by F. A. Coller and W. G. Maddock. (*Annals of Surgery*, 1934, *100*:983.)

[6] Changes in Surface and Rectal Temperatures in Man Produced by the Ingestion of Food Subsequent to Twenty-four Hour Fast, by J. F. Herrick and Charles Sheard. (*American Journal of Physiology*, 1935, *113*:62.)

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results of further investigations regarding the dependence of the dissipation of heat from the body, in the zone of body cooling, on the state of dilatation of the peripheral blood vessels and the relative rôles of the upper and lower extremities in the control of heat elimination under environmental conditions ranging from 18 C to 32 C were previously presented in a paper[7] to this Society. It was pointed out that measurements of the skin temperatures of the fingers and toes serve as the most delicate and sensitive indicators of the changes in the vasomotor tone of the superficial blood vessels, in order that the rate of heat loss may equal the rate of heat production. Hick, Keeton and Glickman[8] extended these investigations through their study of the relationships between skin temperature and effective temperature scale (Houghten and Yagloglou[9]) and found that, in the zone of 66 to 72.5 deg ET, the control of loss of heat by radiation rests largely in the hands and that, in the higher range of 70 to 75 deg ET, it is largely in the feet.

In this contribution are presented: (1) the results of recent investigations on the changes in temperatures of the fingers and toes and the relative rôles of the upper and lower extremities in the dissipation of heat from the body, under various closely controlled environmental conditions which were changed in increments of 1 to 2 C within the range of 18 to 34 C; (2) the relationship between the temperatures of the toes and basal metabolic rates obtained at 25 C and 40 per cent relative humidity; (3) the effect of relative humidity with various atmospheric temperatures on the temperatures of the extremities, and (4) the cooling and warming rates of the toes in subjects with normal circulation. Comparisons of these data are made with the effects produced by increased production of heat and changes of environmental conditions on the skin temperatures of individuals with peripheral vascular disease.

EXPERIMENTAL CRITERIA

In order that significant and reproducible data may be obtained it is of importance that there be satisfactory regulation and control of: (1) the environmental conditions, (2) the person under test, and (3) the physical methods for measurement of skin temperatures. Details regarding the control of environmental temperatures and humidities, thermocouples, and the measurement of skin temperatures are given elsewhere.[10, 11] Air movement was confined to 15 to 30 fpm. The subjects wore light-weight pajamas or shorts and remained in the horizontal position on a comfortable bed during the period from 9 a.m. to 5 p.m. The necessity of an adequate period for adjustment of

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[7] Investigations on the Exchanges of Energy Between the Body and Its Environment, by Charles Sheard, M. M. D. Williams and B. T. Horton. (ASHVE TRANSACTIONS, Vol. 43, 1937, p. 115.)

[8] Physiologic Response of Man to Environmental Temperature, by F. K. Hick, R. W. Keeton and Nathaniel Glickman. (ASHVE TRANSACTIONS, Vol. 44, 1938, p. 145.)

[9] Determination of the Comfort Zone, by F. C. Houghten and C. P. Yagloglou. (ASHVE TRANSACTIONS, Vol. 29, 1923, p. 361.)

[10] The Electromotive Thermometer: An Instrument and a Method for Measuring Intramural, Intravenous, Superficial and Cavity Temperatures, by Charles Sheard. (*American Journal of Clinical Pathology*, 1931, 1:209.)

[11] Loc. Cit. See Note 7.

the body to any given temperature was shown by Freeman and Lindner;[12] hence it is essential that the body remain under any given atmospheric condition for an hour or more, or until fairly constant readings of the extremities (particularly of the toes) are obtained. In an investigation on the effects of posture, Roth, Williams and Sheard[13] found that, under environmental temperatures ranging from 23 to 30 C, with a relative humidity of 40 per cent and the subject in the basal state and free from sweating of the extremities, the skin temperatures of the fingers and toes decreased when the extremities were elevated and increased when the extremities were pendent. Hence the maintenance of the subject in the horizontal position and the movement of bed, subject and thermocouples, as a unit, when it was necessary to change from one to another set of environmental conditions, have contributed greatly to the accuracy of the data which have been obtained.

RELATIONSHIPS OF SKIN TEMPERATURES OF THE NORMAL SUBJECT IN THE BASAL STATE AND UNDER STANDARD ENVIRONMENTAL CONDITIONS

In clinical investigations concerning functional (vasospastic) or organic (occlusive) involvement in either upper or lower extremities (or both) and the changes that may be produced by the administration of typhoid vaccine, by increased production of heat through the ingestion of food or by increases in atmospheric temperature, it is necessary to know the relative values of the superficial skin temperatures, particularly of the fingers and toes, of healthy individuals with normal basal metabolic rates, normal dryness of hands and feet, unclothed and in the horizontal position under certain standard environmental conditions. An atmospheric temperature of 25 C (77 F) and a 40 per cent relative humidity have been found to be satisfactory. Under these conditions, the skin temperatures of the forehead, thorax and extreme upper portions of the arms and legs are commensurate in value and lie within the range of 32 C (89.5 F) to 35 C (95 F), whereas there is a temperature gradient in the extremities, the toes showing the lowest temperature.

A representative set of data is shown in Fig. 1. In general the temperature of the toes will be slightly above room temperature. Ordinarily, the time-temperature relationships of all toes and fingers, *per se*, closely parallel each other, with possible temperature differences ranging between 1 and 2 C. The forehead, thorax and upper part of the legs are 7 to 10 C warmer than the toes and lie approximately within the range of 32 to 34 C. Since the temperature of the skin in any area is the resultant of the heat brought to it largely by its blood supply and the heat dissipated from its surface, there is a marked (but normal) vasoconstriction of the peripheral vessels of the feet, as is shown by the relatively low temperatures of the toes, which is operative to conserve the heat of the body. In contrast to this condition, however, there is

[12] Some Factors Determining the Variability of Skin Temperature, by H. Freeman and F. E. Lindner. (*Archives Internal Medicine*, 1934, *54*:981.)

[13] Changes in the Skin Temperatures of the Extremities Produced by Changes in Posture, by G. M. Roth, M. M. D. Williams and Charles Sheard. (*American Journal of Physiology*, 1938, *124*:161.)

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evidence of considerable vasodilatation of the peripheral vessels of the fingers, and the finer adjustments or regulations of the dissipation of heat are controlled by the fingers and hands, as is shown by the fluctuating temperatures of the fingers.

The temperatures of the toes under standard environmental conditions will be modified by the presence of moisture on the feet and by the rate of heat production. If the extremities are moist, the temperature of the toes may be below room temperature and remain so throughout the course of the day. This effect is doubtless due to the loss of heat by evaporation. This condition can be demonstrated in an individual with dry extremities by placing a film of water of the same temperature as the part of the foot to which it is applied and

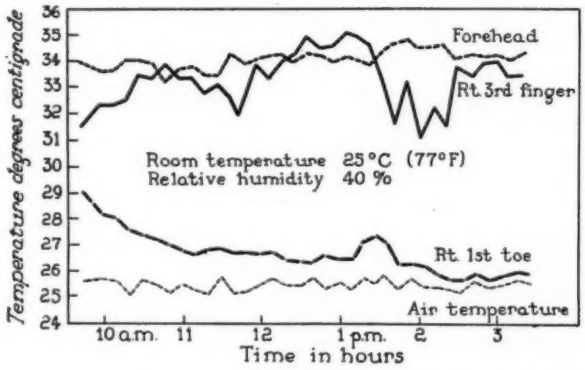

FIG. 1. CURVES OF SKIN TEMPERATURES OF FOREHEAD, FINGER AND TOE OF A NORMAL SUBJECT IN THE BASAL METABOLIC STATE IN AN ENVIRONMENTAL TEMPERATURE OF 77 F (25 C) AND 40 PER CENT RELATIVE HUMIDITY

observing the subsequent reduction in the temperatures of the toes of the extremity involved. Furthermore, all other conditions remaining the same, the drying of a moist foot or hand will be followed by a rise in temperature.

RELATIVE RÔLE OF THE EXTREMITIES IN DISSIPATION OF HEAT UNDER VARIOUS ENVIRONMENTAL CONDITIONS

The relative rôle of the upper and lower extremities, as indicated by measurements of skin temperatures of the fingers and toes, in the regulation of the dissipation of heat from the body has been investigated in several individuals with normal circulation under environmental temperatures which have been increased by increments of 1 to 2 C, the relative humidity remaining 40 per cent. Table 1 presents the data concerning the temperatures of the forehead, upper part of the leg, fingers and toes of two subjects systematically tested under room temperatures ranging from 18 C (64.5 F) to 34 C (93.2 F).

The basal metabolic rates were 40.7 cal per square meter per hour for subject *A* and 36.4 cal per square meter per hour for subject *B*. The range of variation of temperatures for the four areas given in the table occurred in each instance during a period of four to six hours' exposure to the specified atmospheric temperature. The relationships between the temperatures of the toes and fingers and the room temperatures for three subjects of somewhat different basal metabolic rates are shown in Fig. 2.

TABLE 1—SKIN TEMPERATURES OF SELECTED AREAS OF THE BODY OF NORMAL SUBJECTS IN THE BASAL STATE UNDER ENVIRONMENTAL TEMPERATURES MAINTAINED AT VARIOUS DEGREES. (RELATIVE HUMIDITY, 40 PER CENT)[a]

SUBJECT	ROOM TEMPERATURE		RANGE OF SKIN TEMPERATURE, C			
	F	C	Forehead	Leg (Above Knee)	Fingers	Toes
A	67.1	19.5	32.3—32.5	32.1—35.0	31.0—32.5	18.3—18.8
	69.8	21.0	33.0—33.4	30.9—32.0	31.9—35.0	20.4—23.1
	72.5	22.5	33.0—33.8	32.2—33.0	32.8—33.7	24.8—26.7
			32.4—33.6	31.9—32.3	33.9—34.2	22.8—27.2
	77.0	25.0	34.1—34.3	33.5—33.8	33.9—34.0	28.6—29.7
	78.8	26.0	33.3—33.7	33.6—34.0	34.3—36.2	31.4—32.2
	79.7	26.5	34.2—34.5	32.9—33.5	34.0—35.9	32.5—33.3
	83.3	28.5	34.6—35.0		34.6—35.0	34.0—34.7
	93.2	34.0	35.6—36.3	34.8—35.2	34.3—36.3	35.1—35.6
B	64.5	18.0	32.7—33.0	29.4—31.7	24.0—28.1	17.3—17.8
	68.0	20.0	33.5—33.8	30.1—31.7	27.4—29.4	18.1—18.7
	68.9	20.5	34.2—34.7	31.3—32.9	31.0—35.1	19.2—20.1
	69.8	21.0	33.5—33.8	30.7—32.7	28.0—30.5	19.8—20.5
	72.5	22.5	33.6—33.9	31.1—32.8	33.0—34.3	21.0—21.4
	74.3	23.5	33.3—34.2	32.2—32.9	30.0—33.5	21.6—22.3
	76.1	24.5	34.1—34.3	30.9—33.9	33.1—34.7	23.6—29.0
			33.3—34.2	33.0—34.3	31.6—33.7	23.5—25.2
	77.0	25.0	35.0—35.3	32.8—33.6	35.3—36.0	28.4—30.1
	78.8	26.0	33.8—34.3	32.3—34.0	33.0—35.2	30.5—32.6
	84.2	29.0	34.5—35.1	33.4—33.9	34.8—35.7	32.7—33.6
	89.6	32.0	35.8—36.3	34.7—35.6	35.2—36.0	34.3—35.3

[a] The range of skin temperatures for any of the areas specified occurred during a period of observation of from 4 to 6 hours.

Under environmental conditions ranging from 18 to 22 C the temperatures of the toes are, in general, at or near atmospheric temperature, whereas increased vasodilatation is occurring in the upper extremities as is indicated by the rise in temperature of the fingers. When the fingers reach temperatures of 33 to 35 C, indicating approximately maximal normal vasodilatation, the temperatures of the toes exceed the environmental temperature. This ordinarily occurs under a room temperature of 25 to 26 C. Further increase of environmental temperature produces little if any change in the temperatures of the fingers, whereas the temperatures of the toes increase rapidly and reach a value comparable to the temperatures of the fingers under an environmental temperature of 28 C (82.5 F) to 29 C (84 F). Hence, in normally reacting subjects, as the temperature of the room is increased, the regulation of the

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dissipation of heat is accomplished initially by the hands and lower portions of the arms (since normal vasoconstriction of the lower extremities is present). This is followed by a regulatory control on the part of the feet and lower portions of the legs when the room temperature reaches 25 C (77 F). Finally, at higher environmental temperatures (28 to 32 C), the temperatures of the fingers and toes closely approximate the temperature of the forehead, thorax, legs and arms. At atmospheric temperatures exceeding about 31 to 32 C, maximal vasodilatation of peripheral blood vessels will be maintained, and the internal temperature of the body will be kept approximately constant by changes in secretion of sweat in order to balance the heat received from the environment. In turn, passing through the indicated range of environmental

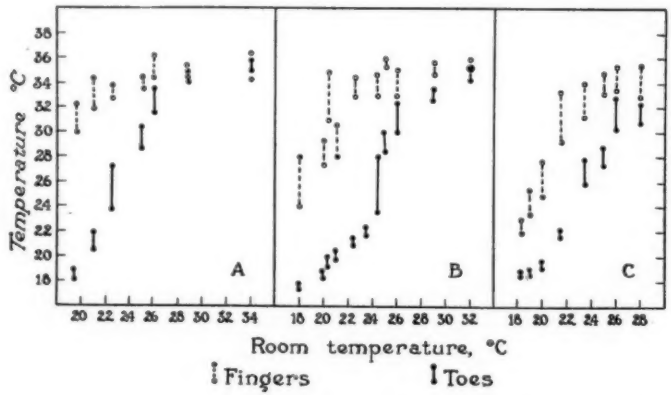

FIG. 2. CURVES SHOWING THE RELATIVE RÔLES OF THE UPPER AND LOWER EXTREMITIES (AS INDICATED BY THE SKIN TEMPERATURES OF THE FINGERS AND TOES) OF NORMAL SUBJECTS UNDER A RANGE OF ENVIRONMENTAL TEMPERATURES EXTENDING FROM 18 TO 34 C

temperatures, in which there is an increased loss of heat due to radiation plus convection and an increased positive storage or cooling of the body tissues (Winslow,[14] et al), and proceeding from the highest to the lowest temperature, it is found that the regulation of the loss of heat is accomplished initially by the lower extremities to the point at which the temperatures of the toes are at room temperature. This indicates normal vasoconstriction of the peripheral vessels of the extremities of the feet and a change in the gradient of vasoconstriction in the lower extremities. Whenever necessary, further regulation of the loss of heat is completed by vasoconstriction of the peripheral vessels of the upper extremities.

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[14] Physiological Reactions and Sensations of Pleasantness Under Varying Atmospheric Conditions, by C.-E. A. Winslow, L. P. Herrington and A. P. Gagge. (ASHVE TRANSACTIONS, Vol. 44, 1938, p. 179.)

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EFFECTS OF CHANGES OF ATMOSPHERIC HUMIDITY ON RESPONSES OF EXTREMITIES

The relative humidity was maintained as near 40 per cent as possible throughout the previous series of tests regarding the relative rôles of the upper and lower extremities in the dissipation of heat from the body, as judged by the temperatures of the fingers and toes. It was of interest, therefore, to determine whether changes in relative humidity at any specified air temperature

TABLE 2—DATA REGARDING THE SKIN TEMPERATURES OF THE FINGERS AND TOES OF SUBJECTS PLACED UNDER SPECIFIED ENVIRONMENTAL TEMPERATURES AND VARIOUS RELATIVE HUMIDITIES

ENVIRONMENTAL CONDITIONS		SUBJECT C		SUBJECT D	
		Average Temperature, C		Average Temperature, C	
Dry-Bulb Temperature, F	Relative Humidity (Average Value) Per Cent	Fingers	Toes	Fingers	Toes
70	53	31.5	22.1	30.2	21.3
	76	28.4	23.3	26.5	22.4
72	40	28.5	22.5	26.5	22.1
	62	30.5	25.6	31.0	23.8
74	40	34.8	30.5	34.5	29.5
	54	32.0	25.4	31.7	24.4
	68	28.0	23.8	27.4	23.6
76	42	34.7	29.5	33.5	28.7
	54	34.0	29.3	33.8	28.8
	73	34.6	29.5	34.3	30.1
79	38	35.1	32.0	35.5	32.5
	50	35.0	32.6	35.1	32.7
	64	35.7	33.8	35.8	34.6
82	34	35.5	33.9	35.6	34.2
	53	35.7	34.3	36.1	34.0
	76	34.9	33.8	35.1	33.6

produced changes in the temperatures of the fingers and toes of two normal subjects with basal metabolic rates of 43.1 and 38.2 cal per square meter per hour. A range of air temperatures from 21 C (70 F) to 27.6 C (82 F) was chosen because the loss of heat due to evaporation is nearly constant and the subjects reported comfort except for some chilliness of feet at the lowest environmental temperature. The temperature of the forehead ranged from 33.2 to 34.6 C. The data recorded in Table 2 show that, within the range of normal variations of environmental and skin temperatures, there is little if any effect of considerable changes in relative humidity at any given atmospheric temperature on the skin temperatures of the fingers and toes. Any minor increases

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in skin temperature which might be produced by large increases in atmospheric humidity are overshadowed by the relatively large thermal changes in the extremities due to small changes in environmental temperature.

THERMAL CHANGES IN THE EXTREMITIES AND INCREASED PRODUCTION OF HEAT

1. *The Ingestion of Food.*—The changes in temperature of the fingers and/or toes and the relative rôles of the upper and lower extremities, under various environmental temperatures, in the dissipation of heat, in the event

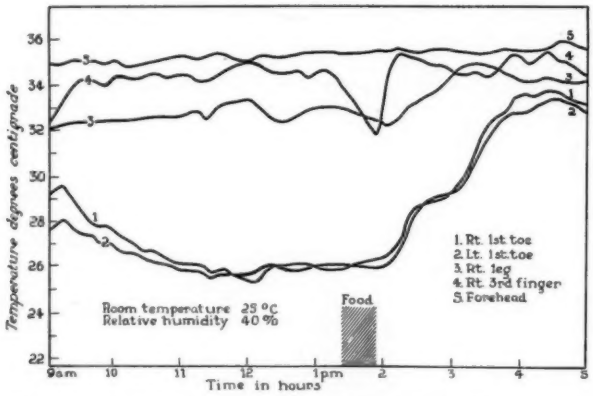

FIG. 3. CURVES SHOWING THE TIME-TEMPERATURE RELATIONSHIP OF THE FOREHEAD, UPPER LEG, FINGERS AND TOES OF A NORMAL SUBJECT BEFORE AND AFTER THE INGESTION OF FOOD. CURVES 1 AND 2, INDICATING THE TEMPERATURES OF THE TOES, SHOW THE RÔLE OF THE LOWER EXTREMITIES IN THE DISSIPATION SUBSEQUENT TO THE INCREASE IN HEAT PRODUCTION WHEN THE SUBJECT IS IN AN ATMOSPHERIC TEMPERATURE OF 25 C

that an increased production of heat occurs, were considered previously. In order that data regarding the changes in skin temperature of a normal subject subsequent to the ingestion of food may be available for comparison with the results obtained for persons with peripheral vascular disease, the records shown in Fig. 3 are included. The data plotted in the left-hand portion (from 9 a.m. to 1 p.m.) show that, in a person in the basal state and in an atmospheric temperature of 25 C, the temperatures of the fingers and forehead were respectively approximately 34 to 35 C, whereas the toes were near room temperature. These skin temperatures are indicators of approximately maximal peripheral vasodilatation in the upper extremities with normal vasoconstriction in the feet and lower part of the legs. Hence the regulation of the elimination of heat should be assumed by the lower extremities in the event that the production of heat is increased and the thermal changes of the toes should vary

somewhat commensurately with changes in the rate of heat production. This is shown by the considerable rise in temperature of the toes (from 26 C to 33 C) in the course of three hours after the ingestion of a standard meal.

2. *Relationship Between Basal Metabolic Rates and Temperatures of the Toes.*—Maddock and Coller [15] concluded that the close connection between the temperature of the extremities and the mechanism for the dissipation of heat is shown by the fact that, under constant environmental conditions, a simple

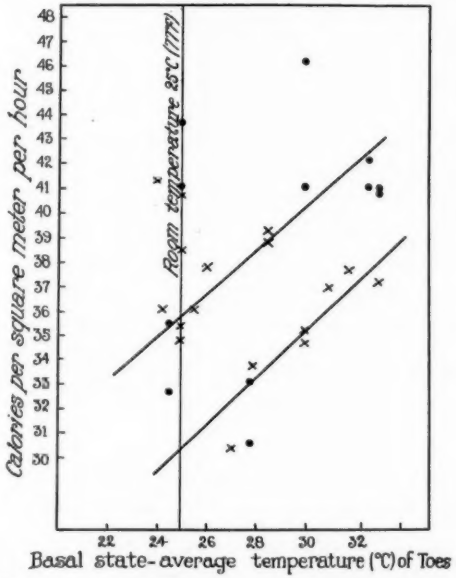

FIG. 4. THE RELATIONSHIPS BETWEEN THE AVERAGE TEMPERATURES OF THE TOES OF A GROUP OF NORMAL SUBJECTS AND THE BASAL METABOLIC RATES. THE ATMOSPHERIC CONDITIONS WERE MAINTAINED CONSTANT AT 25 C AND 40 PER CENT RELATIVE HUMIDITY

linear relationship is found between the temperature of the great toes and the basal heat production per unit of surface area. The experimental data on which this conclusion is based are meager and open to some criticism. The results of the authors' findings for a small group of normal subjects are shown in Fig. 4. There is a rather wide scattering of points, but there is evidence to show that there is an approximately linear relationship of a dual character existing between the average temperatures of the toes and the basal metabolic rates obtained under environmental conditions of 25 C and 40 per cent relative

[15] Loc. Cit. See Notes 4 and 5.

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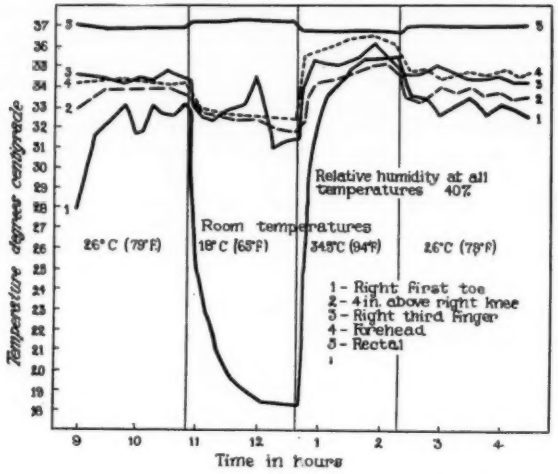
Temperature degrees centigrade
Time in hours
Relative humidity at all temperatures 40%
Room temperatures
26° C (79° F)
18° C (65° F)
34.5° C (94° F)
26° C (79° F)
1 - Right first toe
2 - 4 in. above right knee
3 - Right third finger
4 - Forehead
5 - Rectal

FIG. 5. CURVES SHOWING THE CHANGES IN SKIN TEMPERATURES PRODUCED IN VARIOUS REGIONS OF THE BODY WHEN THE SUBJECT WAS MOVED FROM AN ATMOSPHERE OF 26 C TO COOLER OR WARMER ENVIRONMENTAL TEMPERATURES. CURVE 1 INDICATES THAT THE CHIEF VASO-MOTOR RESPONSES ARE IN THE LOWER EXTREMITIES

humidity. The two lines of linear relationship are roughly parallel to each other. This twofold relationship cannot be explained, since there is no connection between sex (all subjects were males), season of the year, or other factors of this character. However, the type (slender or stout) of the individual and the presence of sweat on the extremities, or normally dry feet and hands as the case may be, may afford some possible explanation.

COOLING AND WARMING RATES OF THE TOES

When the normal subject in the basal state is moved from an environment of 26 C (79 F) to an environment of 18 C (65 F), there is relatively little thermal change in the forehead, thorax, arms and upper portions of the legs. In contrast, there is definite cooling of the toes, and constancy of temperature (approximately that of the room) is reached in about an hour. In turn, when the subject is moved from a room at 18 C to an environment of 34.5 C (94 F), the changes in temperature are again most pronounced in the toes. The rise in temperature is rapid at first; this is followed by a gradual change as the temperature of the toes approximates a maximal value of about 35 C. Curve 1 in Fig. 5 gives representative data regarding the cooling and warming of the toes under the two specified environmental conditions.

If Newton's law of cooling is applicable, then the rate of cooling or warming of the tissue should be proportional to the difference between the temperature of the tissue and its surroundings. The relationship may be written as an exponential function:

$$T_t = T_0 e^{-at}$$

in which T_0 is the initial (zero time) difference in temperature, T_t is the difference in temperature at any specified time, t, and a is the constant. It follows that:

$$a = \frac{1}{t} \log_e \frac{T_0}{T_t}$$

If data such as those given in Fig. 5 are plotted on semi-logarithmic paper, the ordinate being the difference between the temperature of the skin and of the room at any given time, t, indicated as the abscissae, there should be a straight line if the rate is a constant. If there is a change in the rate of cooling or warming of the extremities (toes) of individuals having normal circulation, there should be corresponding changes in the slopes of the lines representing the relationship between the difference of temperatures and the time.

The data plotted in Fig. 6 show that the range of rates of cooling of the toes in normal subjects was between 0.02 and 0.05 C per minute. Similar rates

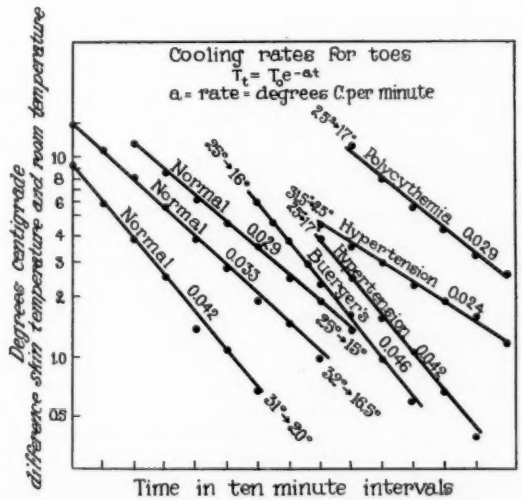

FIG. 6. COOLING RATES OF THE TOES OF INDIVIDUALS WITH NORMAL CIRCULATION AND IN CASES OF PERIPHERAL VASCULAR DISEASE

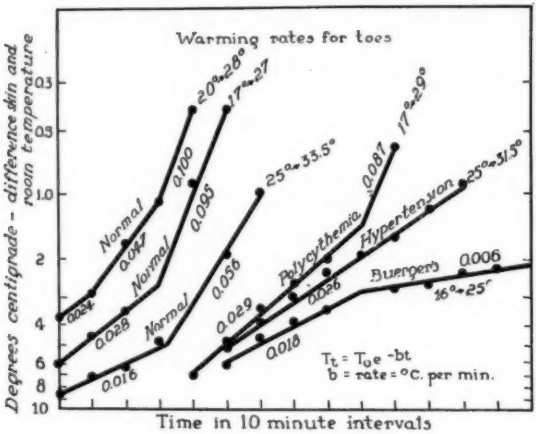

FIG. 7. RATES OF WARMING OF THE TOES OF NORMAL SUBJECTS AND IN CASES OF PERIPHERAL VASCULAR DISEASE

were found for persons with peripheral vascular disease. Since rates of cooling are found to lie within the same range in either normal or abnormal circulatory conditions of the feet, it follows that corresponding changes are present in the supply of blood due to peripheral vasoconstriction and that the same order of change exists in the temperature gradient between the deeper and superficial tissues in all cases. Hence, to the first order of approximation, the rates of cooling of the toes are the same in either normal circulatory conditions or in peripheral vascular diseases, since vasoconstriction, whether spastic or obstructive in character, is operative in all instances.

The data obtained concerning the rates of warming of the toes, as illustrated in Fig. 7, show that there are two (and possibly three) distinct rates falling within the range of 0.015 to 0.15 C per minute. The initial rates of warming lie between 0.015 and 0.030 C per minute and are constant, *per se*, until the temperatures of the toes reach a value of approximately 28 C. A definite change in the rates of warming, ranging in normal circulatory conditions from 0.04 to 0.10 C per minute, occurs at a temperature of about 27 or 28 C. This indicates that there is an increase in the supply of blood to the periphery brought about by vasodilatation of the vessels. Hence there are two distinct values for the rates of warming: the initial value represents the rate of environmental heating of the tissue only, whereas the second rate indicates the combined effects of the heating of tissue by the environment and the increased blood supply. The fact that a change of rate is definitely indicated in the rate of warming at an environmental temperature of about 28 C is in accord with the findings which have been presented regarding the regulatory control of the dissipation of heat, which is assumed initially by the lower extremities when the temperatures of the toes are 24 to 26 C.

If abnormal vasoconstriction (spasm) or organic occlusion is present, the rates of warming of the tissue due to the absorption of heat from the environment will be approximately the same as in normal circulation, but there will be abnormally low or possibly decreased rates of warming in the range of environmental temperature in which vasodilatation normally occurs.

SKIN TEMPERATURES OF EXTREMITIES IN PERIPHERAL VASCULAR DISEASES

Consideration has been given in some detail in the preceding paragraphs to the important rôle of the extremities, as indicated by measurements of the skin temperatures of the fingers and toes of subjects with normal circulation and vasomotor control, in the dissipation of heat from the body under various atmospheric and physiologic conditions. An increase or decrease of superficial temperature is indicative of a corresponding change in the supply of blood to the peripheral tissues. Since changes in the blood supply are controlled chiefly by vasoconstriction or vasodilatation of the blood vessels, it follows that the marked regulation of the dissipation of heat, in the zone of body cooling, is accomplished by an equally marked and varying degree of peripheral vasoconstriction, or vasodilatation as the case may be.

A comparison of the data and information which have been obtained regarding the regulatory function of the extremities in the dissipation of heat with

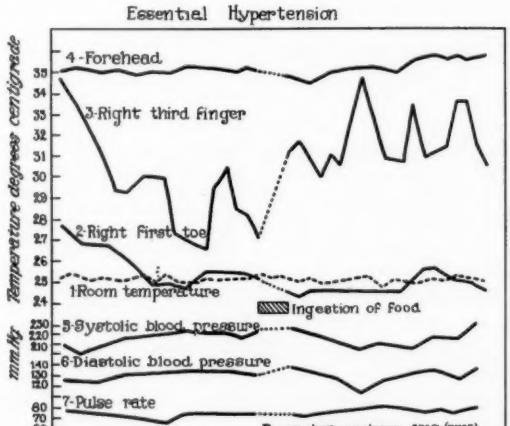

FIG. 8. CURVES SHOWING THE TIME-TEMPERATURE RELATIONSHIPS OF THE FINGERS AND TOES OF AN INDIVIDUAL WITH ESSENTIAL HYPERTENSION. THE TEMPERATURES OF THE FINGERS AND TOES INDICATE ABNORMAL VASOCONSTRICTION IN AN ATMOSPHERE OF 77 F

the findings in peripheral vascular disease is, obviously, of considerable importance to the medical profession from the standpoint of diagnosis and treatment. These data are of signficance to the heating and ventilating engineering profession because of the fact that, under certain environmental conditions, release of abnormal vasoconstriction and a betterment of circulatory conditions may be accomplished, in some measure at least, in many individuals as well as affording added comfort and efficiency.

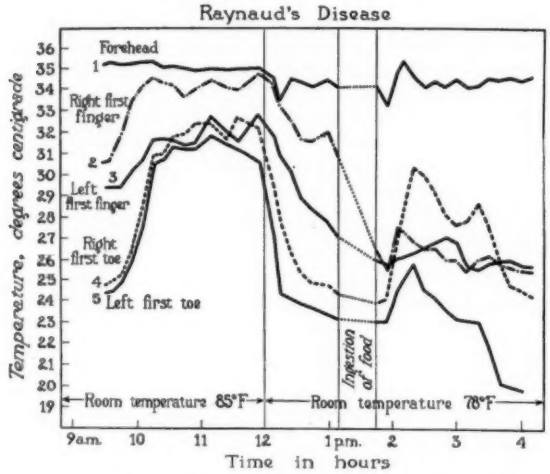

FIG. 9. CURVES SHOWING THE EFFECTS OF ENVIRONMENTAL TEMPERATURE ON THE SKIN TEMPERATURES OF THE TOES AND FINGERS IN RAYNAUD'S DISEASE. THERE IS MARKED VASOCONSTRICTION OF THE FINGERS IN AN ENVIRONMENT OF 78 F WHICH IS REPLACED BY A CONSIDERABLE DEGREE OF VASODILATATION IN AN ENVIRONMENT OF 85 F

In general terms, peripheral vascular deficiencies may be classified in three main groups: (1) organic occlusion only, with little or no spasm, as in advanced cases of thrombo-angiitis obliterans (Buerger's disease); (2) pure spasm with no organic obstruction, as in Raynaud's disease, and (3) some organic occlusion and a variable degree of vasoconstriction. In these individuals, as in normals, the processes of dissipation of heat should occur with a limitation, which is imposed by reason of the inability of the blood vessels to contract or to dilate. The restriction thus imposed may vary considerably in its degree and location in various peripheral vascular diseases. Considerable information regarding the character of the involvement may be acquired from studies of the skin temperatures of the extremities under various environmental conditions.

Sets of data regarding the skin temperatures of the fingers and toes in conditions of essential hypertension, Raynaud's disease and Buerger's disease, are given in Figs. 8, 9 and 10. Since the degree of vasospasm, occlusion, or a combination of both factors varies considerably in different individuals with peripheral vascular disease, the data shown in the curves should be considered only as illustrative of the marked differences in findings which occur in these cases, when compared with the skin temperatures of normal subjects under corresponding environmental conditions.

Essential Hypertension. In the case of an individual with essential hypertension (Fig. 8) and a blood pressure of 220/130 mm of mercury, there is

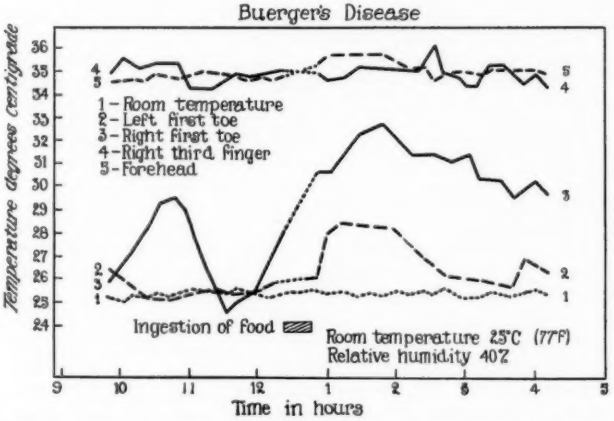

FIG. 10. CURVES SHOWING THE RELATIONSHIP BETWEEN THE SKIN TEMPERATURES OF THE FINGERS AND TOES IN A CASE OF BUERGER'S DISEASE. CURVES 2 AND 3 SHOW THAT INVOLVEMENT IN THE LEFT LOWER EXTREMITY IS GREATER THAN IN THE RIGHT ONE

abnormal vasoconstriction of both extremities as is evidenced by the fact that the skin temperatures of the fingers are low and fluctuate considerably. Following the ingestion of food, the temperatures of the toes remain at room temperature with some rise in, but spasmodic control of, the temperatures of the fingers. In an individual of this type there will be considerable vasodilatation of the extremities when the environmental temperature is raised from 29 to 31 C. Under these atmospheric conditions the temperatures of the fingers approximate those obtained with normal individuals and there will be considerable vasodilatation of the lower extremities, although the temperatures of the toes may be several degrees below the normal findings in corresponding air temperatures.

Raynaud's Disease. As is shown in Fig. 9, the temperatures of the toes are a degree or so above atmospheric temperature (30 C). The temperatures of the fingers, particularly of the left hand, are from 1 to 3 deg below the

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temperature of the forehead. Under similar environmental conditions a normal person exhibits skin temperatures of approximately 35 C for the fingers, toes, thorax and forehead. After placing the subject in an environmental temperature of 25 C, there is a drop in the temperatures of the fingers and toes to values which are much lower than normal. The subsequent ingestion of food shows a rise in the temperatures of the toes, whereas the fingers remain at a temperature slightly above that of the room.

Buerger's Disease. The curves of Fig. 10 show that the temperatures of the fingers and forehead are commensurate (34 to 35 C), whereas the toes are at or slightly below room temperature when in the steady state. Following the ingestion of food there is a rise of 3 C in the temperature of the toes of the left foot as compared with a rise of about 7 C in the toes of the right foot. It is apparent that the degree of involvement is much greater in the left lower extremity. In general, it has been found that the increases in the temperatures of the toes in cases of Buerger's disease following the ingestion of food are practically the same as those obtained after the administration of typhoid vaccine, if allowance is made for the rise of rectal temperature produced by the vaccine.

Vasospasm and Environmental Conditions. As has been suggested in the foregoing considerations, it is possible to determine the environmental temperatures at which the extremities, in cases of peripheral vascular disease, exhibit some degree of vasodilatation. Pickering and Hess [16] stated that they encountered no subjects, normal or diseased, in whom warming the body failed to produce a vasodilator response in the fingers unless the sympathetic nerves were divided or the vessels structurally diseased. Their observations also indicated that warming the body removes only a part of the vasoconstriction tone from the feet, the remainder being more or less permanent. In general, the results of the present investigations are in accord with the conclusions of Pickering and Hess.

In many instances of peripheral vascular disease, the extremities are moist and cool to the touch, and the subjects complain of cold hands or feet, as the case may be. When a marked degree of vasospasm is present, thereby interfering with the normal control of the loss of heat through the vasomotor regulation of the peripheral blood supply, it is probable that the dissipation of heat is accomplished by the evaporation of sweat which is present in abnormal amounts. As a result of the vasospasm and the abnormally high degree of evaporative regulation, which are present under environmental conditions that normally call for dissipation of heat chiefly by the processes of radiation and convection, the temperatures of the toes may be several degrees Centigrade below room temperature (25 C). The temperatures of the fingers may range from normal values (32 to 34 C) to temperatures indicative of pronounced vasoconstriction of the peripheral vessels of the upper extremities. As the atmospheric temperatures are progressively increased, environmental conditions may be found in which considerable dilatation of the peripheral vessels occurs in functional vasospasm. In general, there will be corresponding

[16] Vasodilatation in the Hands and Feet in Response to Warming the Body, by G. W. Pickering and W. Hess. (*Clinical Science*, 1933, *1*:213.)

changes in the temperatures of the toes which are commensurate with the values found in normal subjects, although conditions are encountered in which there is evidence of tissue heating only, with little vasodilatation of blood vessels. In organic occlusion of the lower extremities, with some superimposed spasm, environmental temperatures may be established in which the lower extremities will be normally dry and in which the temperatures of the toes will be above the atmospheric temperature, thereby indicating increased blood supply. This range of atmospheric conditions varies with the degree of vasospasm and occlusion, but may be set at from 28 C (82 F) to 32 C (90 F).

DISCUSSION

DR. M. B. FERDERBER: This paper of Dr. Sheard's and his associates is, perhaps, one of the finest pieces of work on the temperatures of the extremities. We have been doing a little work of our own in Pittsburgh relative to the effects of heat in peripheral vascular diseases. While we have not used rooms, we have used a small enclosure made especially to heat the extremities, and we find that skin temperatures of the extremities increase to within very close to normal. This bears out a statement in the *Journal of the American Medical Association* concerning inquiry regarding what is called passive vascular exercise for vascular diseases. This is nothing more than a pressure-suction device, to give positive and negative pressure to the extremities. One phase is to dilate the blood vessels and the next phase is to push blood in.

We have noticed in Pittsburgh, after a rather careful observation of a few hundred cases, that it has produced virtually little or no change and that perhaps the improvement noted is one which may be partially psychic. I think, secondly, that the patients have been given a tremendous amount of rest, but the physician who wrote this particular reply felt that the giving of typhoid vaccine might be instrumental in raising the temperature of that particular extremity.

We have found that giving typhoid vaccine to some of these individuals, who are ill and probably at ages from the fourth, fifth and sixth decade, might be at times a risky procedure, so that, instead of this, careful moist heating of the extremities would tend to bring temperature up and, coincidental with raising the temperature, it appears as though their symptoms are relieved, at least subjectively.

We used this particular standard: Given a person who has this particular type of involvement, after he has been given several hours of treatment over several weeks, he is then made to walk to the hospital, walk to the fourth floor and immediately the skin temperatures of the limbs are taken with rather surprising results. They approach fairly close to the normal of another individual, with the rather pleasing effect that they have had few, if any, symptoms.

I believe that the work of Dr. Sheard and his colleagues is really revolutionary in that it might destroy the idea or the conception that this passive vascular exercise, or the other vascular exercise, whether produced by an apparatus or change of posture, could relieve the symptoms. I believe that this is a most phenomenal piece of work and I think what little work we have done here, while not carried out as carefully, seems to substantiate just this very thing.

DR. BAYARD T. HORTON: Our clinical experience with the use of the Pavaex machine (alternating suction and pressure apparatus) is in keeping with that referred to by Dr. Ferderber. Like many mechanical devices which are supposed to improve the circulation in the extremities, this type of machine was received very enthusias-

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tically by the medical profession and used extensively in a number of medical centers. Startling reports came out regarding the almost miraculous way in which circulation was restored to the extremities. Unfortunately, we have not been able to verify claims of this type and it is our belief that in the due course of time the alternate pressure and suction apparatus will gradually fall into disuse. We have not been able to note any rise in the surface temperatures of the extremities following the use of this apparatus and if there was any definite increase in the blood flow to the extremities it should be reflected in terms of surface temperature.

From a medical standpoint, I feel that the use of intravenous typhoid vaccine is still our most important therapeutic agent in the treatment of subjects with thromboangiitis obliterans. We do not use it in the treatment of subjects with arteriosclerosis obliterans because of the danger involved in giving it to older subjects. Before deductions can be drawn from the use of any mechanical appliance for the treatment of peripheral vascular disease, it is highly important that an adequate control study be carried out in each instance; this, we have religiously attempted to do in our own work.

No. 1110

SKIN TEMPERATURES OF THE EXTREMITIES AND EFFECTIVE TEMPERATURE

By CHARLES SHEARD, PH.D.,* MARVIN M. D. WILLIAMS, PH.D.** AND BAYARD T. HORTON, M.D.,*** ROCHESTER, MINN.

EFFECTIVE temperature was defined by Houghten and Yaglou [1] as "the temperature and humidity condition, or the heat and moisture condition, of the air which determines the transfer of heat between it and the body." Combinations of temperature, humidity and air movement which induce the same feeling of warmth are designated as thermo-equivalent conditions. From a series of investigations,[2, 3, 4] it was concluded that this scale of thermo-equivalent conditions not only indicates the sensation of warmth but also determines the physiological effects on the body induced by heat or cold.

In a recent report, Hick, Keeton and Glickman [5] concluded that the data which they obtained on the skin temperatures of various regions of the body, of persons very lightly clothed, were most satisfactorily correlated with environmental conditions on the basis of the effective temperature scale or index. Using unclothed subjects, Winslow, Herrington and Gagge [6] found that relative humidity had very little effect on comfort in the zone of body cooling (below

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* Director, Division of Physics and Biophysical Research, The Mayo Foundation and The Mayo Clinic.
** Associate, Division of Physics and Biophysical Research, The Mayo Foundation.
*** Consultant, Division of Medicine, The Mayo Clinic.

[1] Determination of the Comfort Zone, by F. C. Houghten and C. P. Yagloglou. (ASHVE TRANSACTIONS, Vol. 29, 1923, p. 361.)

[2] Cooling Effect on Human Beings Produced by Various Air Velocities, by F. C. Houghten and C. P. Yagloglou. (ASHVE TRANSACTIONS, Vol. 30, 1924, p. 193.)

[3] Effective Temperature with Clothing, by C. P. Yagloglou and W. E. Miller. (ASHVE TRANSACTIONS, Vol. 31, 1925, p. 89.)

[4] Heat Given Up by the Human Body and Its Effect on Heating and Ventilating Problems, by C. P. Yagloglou. (ASHVE TRANSACTIONS, Vol. 30, 1924, p. 365.)

[5] Physiologic Response of Man to Environmental Temperature, by F. K. Hick, R. W. Keeton and Nathaniel Glickman. (ASHVE TRANSACTIONS, Vol. 44, 1938, p. 145.)

[6] Relations Between Atmospheric Conditions, Physiological Reactions and Sensations of Pleasantness, by C.-E. A. Winslow, L. P. Herrington and A. P. Gagge. (*American Journal of Hygiene*, 1937, *26*:103.)

Presented at the 45th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Pittsburgh, Pa., January, 1939

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86 F for the unclothed body), whereas the influence of air movement was very great, particularly with low air temperature. The chief physiological response which these investigators found in the zone of body cooling was an adaptive fall in skin temperature.

In this paper the results of investigations concerning the skin temperatures (chiefly of the fingers and toes) are presented, obtained on two subjects, with normal circulation, unclothed, and in the basal metabolic state, under various environmental conditions which ranged from 66 to 77 deg ET as indicated on

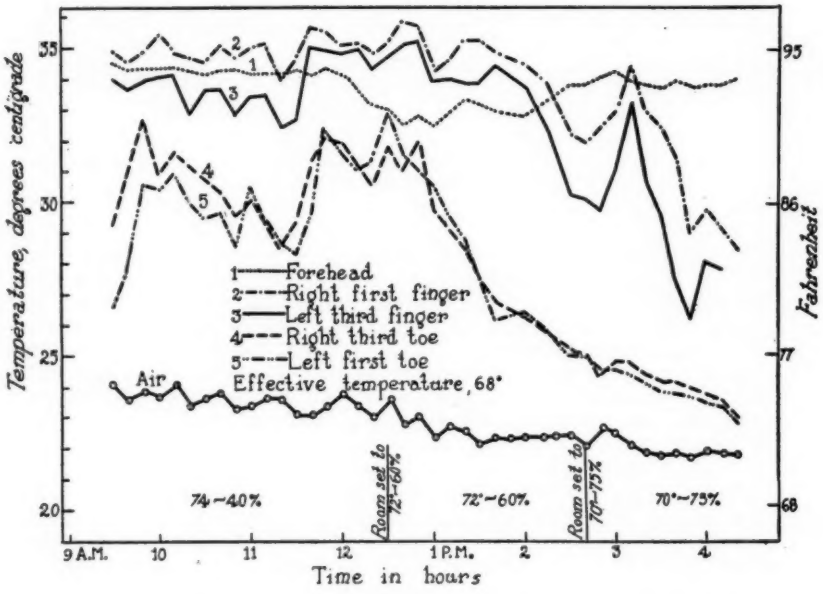

FIG. 1. SKIN TEMPERATURES OF THE FOREHEAD, FINGERS AND TOES OF A NORMAL PERSON UNDER AN EFFECTIVE TEMPERATURE OF 68 DEG

the psychrometric chart [7] for persons at rest, normally clothed and in still air. The skin temperatures of the fingers and toes have been chosen for the reason that, as pointed out in other papers [8, 9] the extremities play an important rôle in the dissipation of heat from the body and for the further reason that thermal

[7] Effective Temperature Index of Warmth. (HEATING, VENTILATING, AIR CONDITIONING GUIDE, 1937, *15*, p. 64.)

[8] Effects of Changes in Environmental Conditions on Skin Temperatures and the Dissipation of Heat from the Body, by Charles Sheard, M. M. D. Williams and B. T. Horton (*American Journal of Physiology*, 1937, *119*:403).

[9] Investigations on the Exchanges of Energy Between the Body and Its Environment, by Charles Sheard, M. M. D. Williams and B. T. Horton (ASHVE TRANSACTIONS, Vol. 43, 1937, p. 115).

changes of the fingers and toes are the most sensitive or delicate indicators of change in the vaso-motor control which, presumably, involves the whole surface of the integument. Furthermore, the changes in the temperatures of the toes are of the greatest significance since, within the range of 66 and 77 deg ET, the temperatures of the forehead, thorax and upper portions of the legs and arms are commensurate and high in value, and the upper extremities (as evidenced by the temperatures of the fingers) are involved but little in the regulation of heat dissipation, whereas definite evidence of vaso-motor

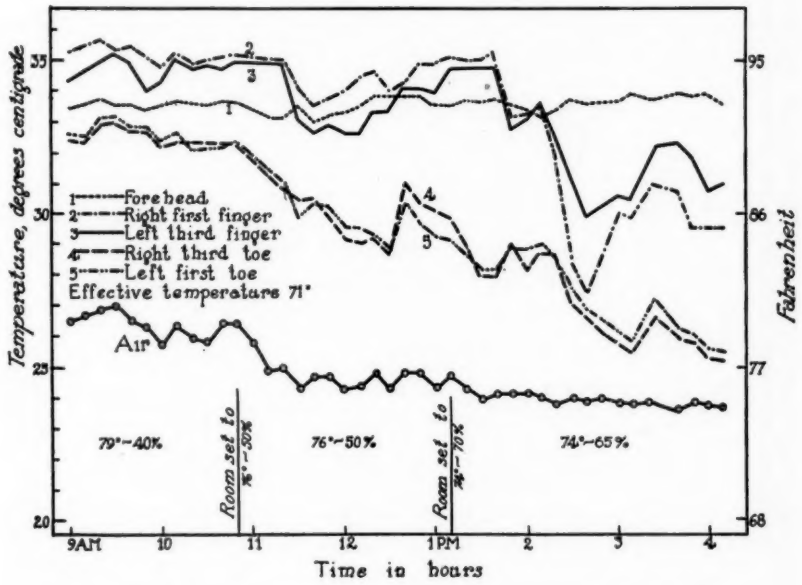

FIG. 2. CURVES SHOWING THE TIME-TEMPERATURE RELATIONSHIPS OF THE FOREHEAD, FINGERS AND TOES UNDER AN EFFECTIVE TEMPERATURE OF 71 DEG

regulation is demonstrated by the spread of the temperatures of the toes from approximately 22 C to 35.8 C.

EXPERIMENTAL RESULTS

Figs. 1, 2 and 3 show the time-temperature relationships obtained on the same person when subjected to atmospheric conditions indicated as 68, 71 and 73 deg ET with an air movement of 15 to 40 fpm.

The data obtained at either a low (66 deg) or high (77 deg) effective

TABLE 1—DATA REGARDING SKIN TEMPERATURES OF THE FINGERS AND TOES, BLOOD PRESSURES AND PULSE RATES OF TWO NORMAL INDIVIDUALS UNDER VARIOUS ATMOSPHERIC CONDITIONS INDICATED BY THE EFFECTIVE TEMPERATURE SCALE

ENVIRONMENTAL CONDITIONS			SUBJECT A				SUBJECT B				COMMENTS ON COMFORT
Effective Temperature	Dry-Bulb Temperature, F	Per Cent Humidity	Average Temperature, C		Blood Pressure (mm Hg.)	Pulse Rate	Average Temperature, C		Blood Pressure (mm Hg.)	Pulse Rate	
			Fingers	Toes			Fingers	Toes			
66	71	40	28.5[a]	22.5	90/60	52	26.5[a]	22.0	94/68	60	Comfortable, except feet cold
	70	55	31.5[a]	22.1	98/64	48	30.0[a]	21.5	94/60	48	Slightly chilly
	69	65	30.2[a]	21.8	88/68	52	29.5[a]	21.7	90/64	50	Slightly chilly, feet cold
68	74	40	34.8	30.5	92/58	56	34.5	29.5	94/60	60	Comfortable
	72	60	30.5	25.6	94/62	54	31.0	23.8	94/66	56	Comfortable, feet cold
	70	75	27.4	23.3	84/68	46	26.5	22.4	94/64	48	Comfortable, feet cold
70	76	40	34.7	29.5	100/74	56	33.5	28.7	96/72	52	Comfortable
	74.5	55	32.0	25.4	88/72	46	31.7	24.4	90/72	50	Fairly comfortable, except feet
	73	65	28.0	23.8	80/60	50	27.4	23.6	90/68	52	Uncomfortably cool
71	79	40	35.1	32.0	96/78	50	35.5	32.5	96/76	66	Very comfortable
	76	50	34.0	29.3	98/70	50	33.8	28.8	98/70	70	Comfortable, slightly cool feet
	74	65	30.5	26.0	90/66	52	32.0	24.5	100/72	60	Uncomfortably cool
73	82	35	35.5	33.9	94/64	51	35.6	34.0	98/70	62	Very comfortable
	79	50	35.0	32.6	90/66	52	35.1	32.7	96/68	64	Very comfortable
	77	70	34.6	29.5	90/60	49	34.3	30.1	96/64	58	Very comfortable, legs and arms slightly moist
75	85	30	35.8	34.6	90/62	51	36.3	33.8	94/60	65	Somewhat uncomfortable
	82	50	35.7	34.3	88/60	64	36.1	34.0	90/62	57	Very comfortable
	79	60	35.7	33.8	92/64	60	35.8	34.6	90/66	52	Comfortable
77	86	40	35.8	34.5	88/66	56	36.0	34.5	104/76	64	Uncomfortable—sweating
	83	60	35.7	34.3	86/64	54	35.7	34.2	106/74	58	Slightly uncomfortable, too warm
	81	73	34.9	32.7	92/70	52	35.1	33.2	102/76	60	Comfortable. Forehead, hands, feet dry

[a]Temperatures of the fingers vacillated considerably, covering a range as great as 6 C. (See Fig. 1.)

Two values of the basal metabolic rate were obtained for each subject, as follows: *A*, −2 (44.5 calories per square meter per hour) and −7 (41.7 calories per square meter per hour); *B*, −10 (39.2 calories per square meter per hour) and −17 (34 calories per square meter per hour).

All data were obtained with the subjects in the basal state, wearing pajamas and lying in the horizontal position.

temperature afford little evidence regarding the equivalence of combinations of dry-bulb and wet-bulb temperatures. The time-temperature relations for the forehead, fingers and toes under an effective temperature of 77 deg show that the skin temperatures of these regions of the body lie within the narrow range of 32.5 to 35.8 C. At 66 deg, the other extreme of the effective temperature scale used in these investigations, the temperatures of the toes remained at room temperature throughout the course of the day, with wide fluctuations

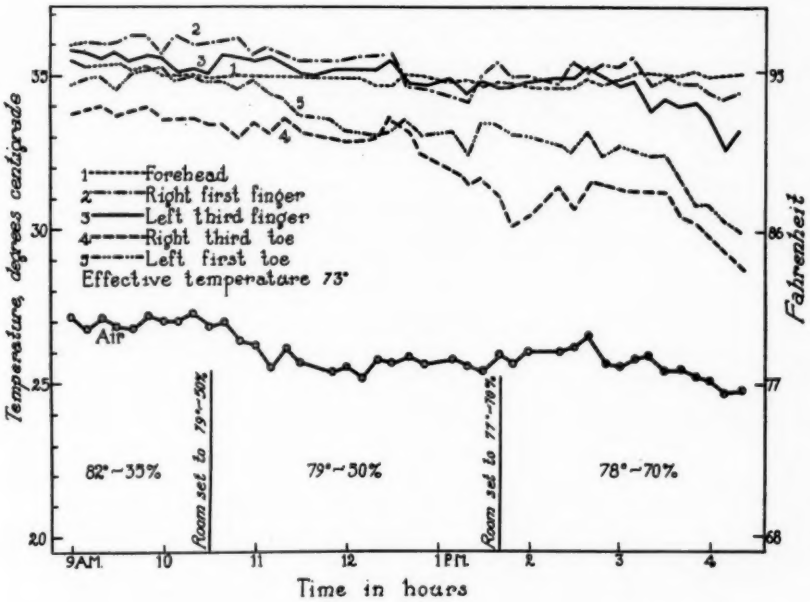

FIG. 3. SKIN TEMPERATURES OF THE FOREHEAD, FINGERS AND TOES OF A NORMAL PERSON UNDER AN EFFECTIVE TEMPERATURE OF 73 DEG

in the temperatures of the fingers. Such results would be expected under the extremes of effective temperature which have been cited, for the reason that, at dry-bulb temperatures of 82 to 86 F, there is maximal peripheral dilatation, whereas, at dry-bulb temperatures of 65 to 75 F, there is normal vasoconstriction of the lower extremities with thermostatic regulation of the dissipation of heat by the upper extremities.

Table 1 contains a résumé of the data regarding physiological responses of two subjects (in the basal state, unclothed, and in the horizontal position) under various effective temperatures. The data show that, within the range of environmental conditions in which the lower extremities play the chief rôle in the regulation of the dissipation of heat, as indicated by the skin tempera-

TABLE 2—CHRONOLOGICAL RECORD OF THE AVERAGE SKIN TEMPERATURES OF THE FOREHEAD, FINGERS AND TOES OBTAINED ON TWO SUBJECTS FOLLOWING AN INITIAL PERIOD OF 15 HOURS' FASTING. DRY-BULB TEMPERATURE 78 F (22.5 C) AND 37 PER CENT RELATIVE HUMIDITY

TIME OF DAY	SUBJECT A			SUBJECT B		
	Average Temperature, C			Average Temperature, C		
	Forehead	Fingers	Toes	Forehead	Fingers	Toes
9:00–11:30 A.M.	34.5	35.3	31.5	35.1	34.9	32.4
11:30– 3:00 P.M.	34.6	35.6	31.0	35.3	34.7	31.6
2:00– 4:30 P.M.	34.8	34.7	29.6	33.7	34.3	29.0

TABLE 3—DATA CONCERNING THE EFFECTS OF VARIOUS ENVIRONMENTAL TEMPERATURES AND RANGES OF RELATIVE HUMIDITY AT ANY DESIGNATED DRY-BULB READING ON THE TEMPERATURES OF THE FOREHEAD, FINGERS AND TOES[a]

ENVIRONMENTAL CONDITIONS		SUBJECT A AVERAGE TEMPERATURE, C			SUBJECT B AVERAGE TEMPERATURE, C		
Dry-Bulb Temperature	Relative Humidity (Average Value) Per Cent	Forehead	Fingers	Toes	Forehead	Fingers	Toes
73 F (22.8 C)	38	33.8	31.0	24.5	34.7	32.0	23.8
	53	33.5	29.5	23.4	34.5	31.0	23.2
	68	34.2	28.5	23.6	35.2	30.5	23.4
75 F (23.9 C)	41	34.2	35.5	32.5	35.6	34.5	32.0
	54	34.3	34.7	31.6	35.2	35.0	30.8
	64	34.7	33.8	29.6	35.0	33.8	28.4
77 F (25 C)	39	33.6	34.5	29.3	33.5	34.0	29.8
	58	34.1	33.8	28.5	34.5	33.5	27.2
	70	34.5	32.5	27.3	34.8	31.8	26.8
79 F (26.1 C)	32	34.5	35.5	32.2	34.5	34.6	32.5
	50	34.8	35.5	31.5	34.8	34.7	31.6
	64	35.3	34.3	29.8	35.2	35.0	31.1
82 F (27.2 C)	30	34.5	34.6	33.6	35.3	35.0	33.2
	53	35.1	35.0	32.8	35.0	35.5	33.5
	66	35.2	35.0	32.5	35.4	35.4	33.4

[a] Basal metabolic rate (taken at dry-bulb temperature of 77 F and relative humidity 40 per cent); subject *A*, −2 (44.5 calories per square meter per hour); subject *B*, −10 (39.2 calories per square meter per hour).

The data tabulated for any given dry-bulb temperature and range of relative humidities were obtained on the same day on both subjects. The two subjects remained under any given environmental temperature and relative humidity for a period of 2 to 3 hours.

tures of the toes, there is little effect of considerable changes in the wet-bulb temperatures in contradistinction to the definite influence of relatively small changes in dry-bulb temperature. At an effective temperature of 71 deg, as an illustrative citation, the average temperature of the toes is 26 C (Subject A) when the air temperature is 74 F with a relative humidity of 65 per cent, and 32 C when the air temperature is 79 F with a relative humidity of 35 to 40 per cent.

The question may be raised concerning effects which might be caused by fatigue and minor changes in the production of heat during the course of the day. Figs. 1 to 3 and the data of Table 1 show that three changes were made in the environmental temperature and relative humidity during a period of seven or eight hours. The order of procedure was to decrease the temperature and increase the relative humidity, for any specified effective temperature, at the end of a two-hour period. Hence the most marked physiological effects, as indicated chiefly by a decrease in the skin temperatures, would occur in all likelihood in the middle to the close of the afternoon and at the time when relatively lower dry-bulb temperatures and higher humidities were maintained. The data of Table 2 indicate that, under an atmospheric temperature of 78 F and 37 per cent (average) relative humidity, there were minor changes in the temperatures of the forehead and fingers of both persons tested, whereas decreases of 1.9 C (Subject A) and 3.4 C (Subject B) were obtained in the temperatures of the toes during the course of the day. The greatest change in the skin temperatures occurred from 2 p.m. to the conclusion of the test. The same order and approximate changes in the daily range of skin temperatures were obtained at 77 F and 60 per cent relative humidity. The application of such correction factors, as are indicated in Table 2, to the data of Table 1 or to the curves of Figs. 1 to 3, does not appreciably affect the results or invalidate the conclusions to be drawn concerning the effects of various thermo-equivalent conditions on the dissipation of heat from the body.

The effects of changes in relative humidity on skin temperatures were investigated further at dry-bulb temperatures ranging from 73 F to 82 F. The two persons who were used in obtaining the data of Table 1 were tested again in a manner which did not involve a predetermined program. During the course of various tests conducted over a period of two months data were accumulated concerning the skin temperatures of various areas of the body when subjected to various environmental temperatures and ranges of relative humidity at any designated dry-bulb reading. The data of Table 3 show that there is little if any effect produced on the skin temperatures by changes in the relative humidity at any designated atmospheric temperature.

CONCLUSIONS

These investigations show that relative humidity has little effect on the skin temperatures of the body, as indicated most sensitively by changes in the temperatures of the fingers and toes, when the body is subjected to various effective temperatures extending from 66 to 77 deg or to fixed environmental temperatures (73 to 82 F) with a fairly wide range (35 to 70 per cent) of

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relative humidity. It is the authors' conclusion, therefore, that, in the zones of vaso-motor regulation and body cooling, the dissipation of heat from the body is dependent chiefly on the environmental temperature and is little influenced by relative humidity, when the person under test is in the basal metabolic state, unclothed and at rest in still air.

No. 1111

AIR CONDITIONING REQUIREMENTS OF AN OPERATING ROOM AND RECOVERY WARD

By F. C. HOUGHTEN * (*MEMBER*) AND W. LEIGH COOK, JR., M.D.,** PITTSBURGH, PA.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the Department of Industrial Hygiene, School of Medicine, University of Pittsburgh, Bindley Fund and the Elizabeth Steel Magee Hospital.

CONSIDERABLE interest has recently been shown in the application of air conditioning in hospitals, especially from a purely comfort point of view. Particular emphasis has been placed upon summer cooling and air conditioning in wards, private rooms, and various other hospital rooms. Temperature control for the air conditioned space is undoubtedly the most important factor, and it is natural that practices have followed closely those used in air conditioning of other places of assemblage.

More recently, there has been an intense interest in special applications of air conditioning to operating rooms, recovery rooms, and contagious wards. While the greater efficiency of the operating surgeon and his staff is an important consideration in such applications, of more interest has been the possibility of reducing spread of infection, arising either in connection with surgery or from contagious cases. This interest has received its major impetus from the re-opening of the question of air-borne infection by the recent work of Wells and others.[1-6] Previously for the past 10 or 15 years it had largely

* Director, Research Laboratory, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.
** Dept. of Industrial Hygiene, School of Medicine, University of Pittsburgh.

[1] Infection in Clean Operating Rooms, by F. L. Meleney. (*Surgery, Gynecology and Obstetrics*, 60:264; 1935.)

[2] Air-Borne Infection, by W. F. Wells and M. W. Wells. (*Journal of American Medical Association*, 107, 21:1698, Nov. 21, 1936.)

[3] Air-Borne Infection: Sanitary Control, by W. F. Wells and M. W. Wells. (*Journal of American Medical Association*, 107, 22:1805, Nov. 28. 1936.)

[4] Sterilization of Air in Operating Rooms, by Special Bactericidal Radiant Energy, by D. Hart. (*Journal of Thoracic Surgery*, 6:45; 1936.)

[5] Ventilation and Atmospheric Pollution: A Bacteriological Method of Sanitary Air Analyses. (Report by Sub-Committee on Bacteriological Procedures in Air Analysis in Year Book, 1936-1937. Supplement to *American Journal of Public Health*, Vol. 27, No. 3, 1937.)

[6] Incidence of Air Borne Bacteria in Major Surgery of Multnomah County Hospital, by Nesbit and Brook. (*Surgery*, November, 1938.)

Presented at the 45th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Pittsburgh, Pa., January, 1939.

been the accepted opinion of the biologist and bacteriologist that the spread of infection resulting from bacteria-infected spray thrown from one person to the other by coughing and sneezing, could not and actually was not spread through the air for distances greater than a few feet.

However, the recent work of Wells has placed an entirely new complexion on this situation. His discovery that bacteria, including pathogenic organisms, can be and are carried through the air long distances and over long periods of time is perhaps today the subject of outstanding new interest in sanitation. As a consequence, the surgeon is actively considering the possibility that what has been accepted as an irreducible minimum of infections in surgical cases has actually been resulting from an entirely unsuspected source; namely, air-borne organisms. The older accepted theory that organisms, particularly pathogenic ones, were not normally carried through the air on dust particles, excepting where considerable air motion was set up, resulted in tightly closed and in most cases unventilated operating rooms. Windows were not opened for fear that drafts would stir up dusts. This school of thought was even carried in many cases to the point where a glass barrier was installed between the operating room and the conventionally sealed windows, in order to avoid small stray air currents and drafts coming through window cracks. This minimum introduction or change of air for ventilation resulted in two major problems: first, very high temperatures during the summer season, resulting from the high outside temperatures and accentuated by such other necessary sources of heat as steam and hot water baths and sterilizers; and second, attendant high concentration of volatile gases used in anesthesia, resulting from the elimination of all sources of air change or air ventilation. This latter condition has existed to such an extent that it has sometimes been said that a surgeon may cut from 10 to 15 years off his normal life expectancy due to continuous inhalation of relatively high concentrations of anesthetic.

These developments led to a consideration of the need of a thorough research investigation of the factors involved in and the desirability for the air conditioning of operating rooms and recovery wards. As a result, a program of investigation was initiated and some phases of it have since been accomplished.

While the possibilities for research were almost unlimited, a program was outlined for the purpose of indicating the most obvious direction of research investigation. These items will be further enlarged upon, including some details of the method of attack, particularly as pertains to the more physical aspects of the application of air conditioning. Item 5, dealing with the physiological and medical aspects, only covers the subject in a very general way and will naturally unfold and be developed as the work proceeds. The program includes a study of the items given:

1. *Feeling of Warmth:* Optimum temperature and humidity conditions within the operating room and recovery ward for the best interests of the patient, either directly, or through greater efficiency of the operating surgeon and his attendants.

2. *Bacteria:* The bacteria content of the atmosphere in an operating room as found in the past, and with the present air conditioning system; and further, how the bacteria content may be affected by ultra-violet sterilization and electrostatic precipitation of dust.

3. *Anesthetic Concentrations:* The relation between past performance without air conditioning and present performance with air conditioning and controlled ventilation, as affecting concentration of anesthetic at different points in the operating room.

4. *Static-Electric Sparks:* Relation between atmospheric conditions and the hazard of electrical sparking possibly causing ignition of air-gas mixtures.

5. *Physiological Aspects of the Subject:* Medical observations relative to weight, blood pressures, blood chemistries, and skin temperature; perspiration of attendants and patients; feeling of comfort, well-being, and presence or absence of fatigue of operators; urine chloride of patient before and after operation; bacteriologic content of operating room air without and with occupants, and effect on healing; and the effect of the recovery room as it aids or deters patient's recovery, which might be contrasted to a room in which no air conditioning exists.

The study indicated in Item 1 has been largely completed and will be discussed at length later in this report. However, this paper deals practically entirely with the engineering aspects of the study as related to the comfort and well-being of the operating surgeon and his attendants. The hospital and medical aspects of the problem will be presented in other publications.

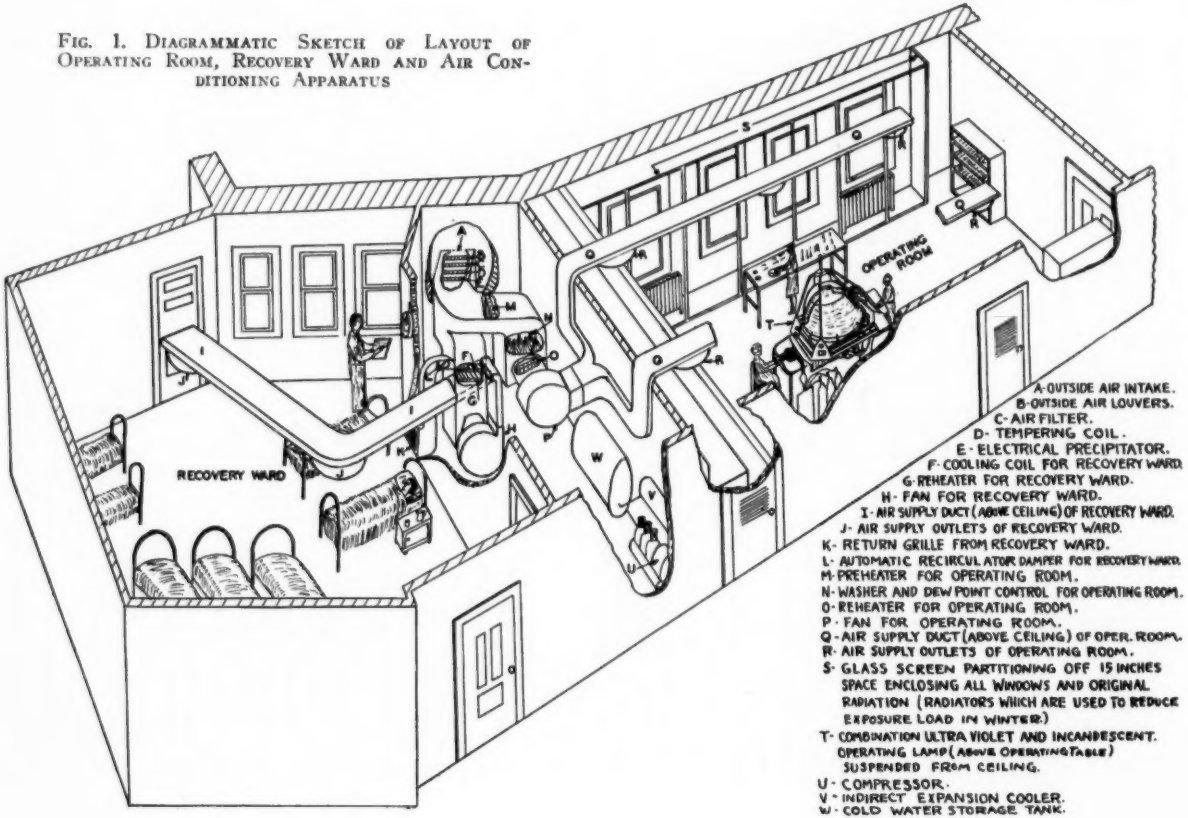
The study of the bacteria content of the air, or Item 2 of the program, has been the subject of considerable difference of opinion. While some bacteria may not be harmful in other localities, as far as an operating room is concerned the presence of undesirable organisms may be considered as roughly proportional to the concentration of all such organisms. Therefore, an approved method of sampling air for bacteria content should be used to sample the air at various points in the operating room under different conditions, both as regards the use of the operating room itself, and the operation of the air conditioning system. Two general methods have been used for this purpose. The earlier procedure was to set sterile petri dishes at various points in the room and allow the bacteria and dust in the atmosphere to settle in them over a period of time and then to develop the bacteria collected. The second, more recent, and probably better method has been to centrifuge the air in a container, the inside of which is covered with an adhesive medium. This method of sampling is positive and quantitative, since a definite volume of air is centrifuged with the idea of getting all or at least a proportionate number of the living organisms out of the air tested. Such an instrument has been developed by W. F. Wells of the University of Pennsylvania.

As far as the percentage of bacteria in the air and their effect are concerned, it is recommended that a few tests under different atmospheric conditions and surrounding arrangements be made, such as without the air conditioning system in operation, with the system in operation including the electrostatic precipitator, and with all of this equipment and the ultra-violet sterilizer also in use. Undoubtedly, a wide variety of different tests should be made, as well as a considerable number of tests for each variation, in order to definitely establish the relation of bacteria content to such operations, the need of wearing masks, etc. Another study would be to determine the desirability of installing a glass barrier, between the operating room and the spectators' gallery, as affecting the bacteria content of the air.

After carefully considering these methods for studying bacteria, a very definite schedule should be set up, aimed to give the greatest amount of information with the least amount of study. The actual incubation of bacteria samples and method to be used thereby must necessarily be dictated by bacteriological experts. Up to date, relatively few observations have been made in connection with this phase of the study.

The concentration of anesthetic in the operating room, or Item 3 of the

FIG. 1. DIAGRAMMATIC SKETCH OF LAYOUT OF OPERATING ROOM, RECOVERY WARD AND AIR CONDITIONING APPARATUS


RECOVERY WARD
OPERATING ROOM
A-OUTSIDE AIR INTAKE.
B-OUTSIDE AIR LOUVERS.
C-AIR FILTER.
D-TEMPERING COIL.
E-ELECTRICAL PRECIPITATOR.
F-COOLING COIL FOR RECOVERY WARD.
G-REHEATER FOR RECOVERY WARD.
H-FAN FOR RECOVERY WARD.
I-AIR SUPPLY DUCT (ABOVE CEILING) OF RECOVERY WARD.
J-AIR SUPPLY OUTLETS OF RECOVERY WARD.
K-RETURN GRILLE FROM RECOVERY WARD.
L-AUTOMATIC RECIRCULATOR DAMPER FOR RECOVERY WARD.
M-PREHEATER FOR OPERATING ROOM.
N-WASHER AND DEW POINT CONTROL FOR OPERATING ROOM.
O-REHEATER FOR OPERATING ROOM.
P-FAN FOR OPERATING ROOM.
Q-AIR SUPPLY DUCT (ABOVE CEILING) OF OPER. ROOM.
R-AIR SUPPLY OUTLETS OF OPERATING ROOM.
S-GLASS SCREEN PARTITIONING OFF 15 INCHES SPACE ENCLOSING ALL WINDOWS AND ORIGINAL RADIATION (RADIATORS WHICH ARE USED TO REDUCE EXPOSURE LOAD IN WINTER.)
T-COMBINATION ULTRA VIOLET AND INCANDESCENT. OPERATING LAMP (ABOVE OPERATING TABLE) SUSPENDED FROM CEILING.
U-COMPRESSOR.
V-INDIRECT EXPANSION COOLER.
W-COLD WATER STORAGE TANK.

program, also should be given serious consideration. As mentioned previously, the life expectancy of surgeons is thought to be shortened an appreciable amount when exposed to high concentrations. Whether this is true or not, nevertheless high concentrations of anesthetic throughout a room are not a necessary consequence to the administering of anesthetic to a patient. Given a certain concentration as necessary at the mouth or nose, the concentration at points distant therefrom can be reduced to almost any desired value by the volume of air change. Of course, there must necessarily be some limitation in the amount of air change or ventilation which can be allowed without undesirable drafts, but it is probable that air change can lower the concentration of anesthetic in the operating room to levels far below those found in usual practice, where no special consideration has been given.

One important phase of the study should therefore be the sampling of air at various points in the room with reference to the source of distribution of anesthetic to the atmosphere, in order to determine the concentration with different rates of air change or different operations of the air conditioning system. The sampling of air is well established by gas chemistry, and analyses for the more common anesthetics used are also available in most instances.

Item 4 of the program, or the relation between atmospheric conditions and the hazard of electrical sparking, and thereby ignition of air-gas mixtures, seems to be of some significance, but to what extent it is important is better known by the operating surgeon and others. Before a study is to be made of this subject, careful analysis should be made first of the available literature on the relation between static electrical sparks and atmospheric conditions in other industries, such as the textile industry. Any future planning of this phase of the subject will have to be developed after due consideration to its importance.

DESCRIPTION OF EQUIPMENT

An operating room and recovery ward at the Magee Hospital, each having a northern exposure on the fourth floor, were chosen for the study and were remodeled in a number of respects, including the installation of an air conditioning system for the rooms. The air conditioning system installed was designed by a committee of doctors and air conditioning engineers, not with a view of fulfilling a predetermined opinion of what air conditioning should apply, but with a view of making available an experimental system which could be operated so as to supply a wide range of atmospheric conditions for experimental purposes.

The layout of the operating room, recovery ward, and air conditioning system is shown in Fig. 1. Outside air is brought in at *A* through a duct containing a preheater, a cloth air filter, and an electrical precipitator. The duct then branches, and part of the air is drawn through an air washer and reheater by a fan, which discharges this air through a duct system to 6 outlets located in the ceiling of the operating room. The second portion of the air, which may be mixed with recirculated air from the recovery ward, is drawn over cooling coils and a reheater by a second fan, which then delivers this air to a duct system leading to two outlets in the ceiling of the recovery

ward. The system is designed to supply 1200 cfm to the operating room and a similar amount, including up to 50 per cent recirculated air, to the recovery ward.

A 10-ton dichlorodifluoromethane compressor applies refrigeration to water by means of a shell-and-tube exchanger. A system of storage tanks, pumps, and thermostatic controls circulates this water through the washer, thus providing cooling and humidity control for the operating room, and also through the cooling coils, thus supplying cooling and dehumidification for the recovery ward. This system therefore allows easy control over both the temperature and dew-point of the air supplied to the operating room, as well as the temperature and, to a lesser extent, the moisture content of the air in the recovery ward.

The operating room is 15 by 32 ft, including a spectators' gallery having a seating capacity for 30 persons. The four operating room windows, with two radiators located under them, are shielded from the rest of the operating room by a glass enclosure. Originally in laying out the system, provision was made for exhausting the air through this space so as to decrease the cooling load from the exposed windows. However, this distribution of the vitiated air from the operating room was not used, for it was soon learned by other attendants in this wing of the fourth floor that the discharged air from the operating room had sufficient residual cooling capacity to add considerable comfort cooling to the rest of the hospital wing when it was discharged into the corridor.

As a part of the experimental equipment in the operating room, an ultraviolet sterilamp is installed directly over the operating table. The recovery ward has a capacity of 7 beds, and has been used solely as a post-operative room.

The experimental equipment was installed and put into operation during June, 1938; and the operating room and recovery ward have since been used continuously, an average of 23 operations being performed there weekly. A resident physician having a few other duties was assigned to the operation of the equipment and collection of the necessary data in connection with the use of the air conditioning equipment for the operating room and recovery ward.

A previous report[7] has recommended that a relative humidity between 50 and 60 per cent be maintained in an operating room, in order to eliminate explosion hazards arising from static electricity. In accordance with this recommendation, all tests in the present study were conducted with various effective temperatures, but all at a relative humidity of approximately 50 per cent.

The study so far has been confined largely to Item I of the research program outlined, and the research results contained in this paper are limited largely to this phase of the study.

EXPERIMENTAL OBSERVATIONS

Coincidental data taken included the types of work done in the operating room, the subsequent course of the patient in the hospital, and the general

[7] Report of Committee on Anaesthesia. (*Journal of American Medical Association*, May 10, 1930.)

reactions of both patient and surgeon toward the equipment in use. An attempt was made throughout to be as purely objective as possible in the gathering of facts and in questions asked. At the outset there was such a general enthusiasm among the operating personnel concerning the cooled space and a willingness to perform elective surgery in the heat of summer, because of the standard conditions which could be maintained for the benefit of the patient, that an abundance of facts confronted the tabulator. Numerous consultations determined the type of information which should be gathered in an effort to have a uniform picture of the work throughout its entirety.

The following problems were suggested for solution concerning the patient's course in the hospital:

1. Do patients who are subjected to a standard avertin, gas, or ether anesthesia develop post-operative pulmonary complications less or in greater percentage when taken directly from the operating room to the air conditioned recovery ward?
2. Do such patients, in a room with 12 air changes per hour, react from anesthesias more quickly than those in humid, sultry wards with little air change and circulation?
3. Is there any perceptible difference in post-operative temperature, pulse, and respiration comparing those kept at a constant condition of approximately 75 F dry-bulb and 40 to 50 per cent relative humidity, and those control cases in the warmer unconditioned wards?
4. Determine the amount of post-operative perspiration, liquid intake, and urinary output in an effort to objectively ascertain the benefit to the patient of being in an ideal post-operative atmosphere.
5. Obtain reactions from both patients and attending physicians as to the advantages or disadvantages of air conditioned recovery wards during the summer months.

During the test months of July, August, and September, 123 patients of the total number operated upon during the summer were placed in the air conditioned recovery ward. Of these, 88 had been operated upon in the air conditioned operating room, and the other 35 had been in ordinary operating rooms. These cases included 36 laparotomies (an operation in which the abdomen is opened), 20 vaginal operations, 48 combined vaginal and abdominal operations, 12 other surgical procedures, and 7 obstetrical cases mostly Caesarean sections. The 123 surgical operations mentioned covered the ordinary types of surgical procedures in a general hospital. The length of stay in the air conditioned recovery ward varied depending upon the need for beds and the general post-operative condition of the patient, the average time being 80 hours and the longest being 9 days.

Throughout the past summer there were but three post-operative complications. One patient died post-operatively of a kidney complication, the patient being 62 years old and a bad operative risk. Another died 6 days after a Caesarean section, the cause being peritonitis. A third patient developed a lung abscess post-operatively which was believed secondary to an embolus. However, none of these 123 cases developed post-operative pulmonary complications. Taking charts from a similar period during the year 1937 it was found that there had been a total of 4 post-operative complications including one death, one case in which the operation reactivated an old tuberculosis, and two cases of post-operative atelectasis (collapse of lung), both of which

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recovered, but were definitely classified as post-operative pulmonary complications.

Comparing the charts of those 123 patients in the air conditioned recovery ward and those others returned to private rooms and the regular surgical wards, no conclusions can be reached as to the advantage of the controlled air condition and air change in reducing the total time required for the patient to react from anesthesia. Because of the rather general use at the Magee Hospital of rectal anesthesia avertin, which is not eliminated through the respiratory tract, it is difficult to determine whether proper air conditioning in the recovery ward is of advantage in reducing anesthesia concentration in the lungs. Pulse and respiration records of patients in the recovery ward and control cases had such wide variations that it would be unable to draw any satisfactory conclusions as to the advantages of the recovery ward post-operatively in these respects, unless a large amount of additional data were to be obtained. The individual reaction of patients to operations and the varying degrees of shock produced by surgical procedures, probably account somewhat for the inability to show any objective advantage of an air conditioned recovery ward.

From a medical standpoint there was some debate as to the advisability of subjecting an anesthetized patient to even the slight ranges of heat and cold used in collecting the data for this paper. This feeling was founded on the statements of Barber and Bourne [8] regarding ether anesthesia on dogs, and the generally accepted but hardly proven theory that anesthesia causes the subject to lose heat control regulation. Rectal temperatures were taken throughout laparotomies of duration of 1 to 1½ hours to see the effect of avertin, gas, or ether anesthesia on the body temperature. Five recordings were made during the operations for chronic pelvic complaints, not associated with pyrexia, and two pre-operative control recordings were made for periods of 2 hours at a time. In no case was there a variation of rectal temperature of more than ⅘ F. Three of these recordings were made in a room conditioned to a temperature of 75 F and a humidity of 50 to 55 per cent, and the others were in rooms well over 80 F with varying humidity, not conditioned. It was therefore thought that with the patient properly draped he could be kept at a safe body temperature even though the conditions of the room were varied.

A careful record of liquid intake and output, which was done in an attempt to evaluate roughly the loss of fluid from the skin and respiratory tract, was made on those patients in the air conditioned recovery ward and controlled cases throughout the hospital. It was thought that the patients in the recovery ward would have a higher urine output in comparison to liquid intake than those in the warmer parts of the hospital, thereby proving less post-operative perspiration. Such, however, was not the case, the variations of intake and output being as great in the air conditioned recovery ward as with the controlled cases. However, mention in the nurses' notes was made by those working in both conditioned and non-conditioned wards that there was much less need to change gowns post-operatively on those cases kept in an ideal post-operative condition, and that there was much less restlessness on the part of the patient in these conditions.

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[8] The Influence of Ether in Dogs, by H. G. Barber and W. Bourne. (*American Journal of Physiology*, 67:399, 1924.)

Numerous operations which would likely be delayed until cooler weather were performed throughout the summer by the surgical staff. The advantages of the air conditioned operating room and recovery ward in lessening the possibility of post-operative heat prostration in the extremes of summer heat undoubtedly influenced their decisions.[9] Operations on the thyroid gland which are more hazardous in the summer time were performed with very little if any apprehension on the part of the staff, and no post-operative thyroid crises resulted during this period.

Throughout the summer, patients continually remarked about the difference in their general feeling of well-being in the air conditioned recovery ward as compared with that on transfer to the unconditioned part of the hospital, even though they were nearer full recovery from their operation when transferred. Other patients who had had previous operations remarked that they had never been so comfortable post-operatively; naturally many were miserable enough to be entirely ignorant of their ideal surroundings.

Also, numerous comments from the medical staff about the air conditioned operating room were heard throughout the summer. These remarks were to the effect that following a whole morning spent in an operating room the surgeon felt he could go to his office with a freshness never before experienced during the hot summer months. Many felt that the air conditioned operating room was far superior to the ordinary operating room regarding comfort, freedom of perspiration and lack of fatigue. The fact that members of the surgeon's family and his patient noticed to a much less extent the odor of anesthesia on the surgeon's breath after a morning in the air conditioned operating room was also mentioned by some of the staff.

COMFORT REACTIONS

Data were collected throughout the summer months on Item 1 of the original program, or the study of optimum conditions of effective temperature for the comfort of the surgeon and those in attendance. With the necessary modifications dictated by the normal operation of the hospital, the study was directed along lines similar to those used by the ASHVE Research Laboratory in earlier comfort studies,[10, 11, 12, 13]. On a given day, a predetermined atmospheric condition, as determined by the relative humidity and effective temperature, was maintained. On convenient occasions an observer interrogated the operating surgeon and his attendants, including internes and nurses, concerning their feeling of warmth, degree of sensible perspiration, and general reactions toward the atmospheric conditions. A feeling of warmth was graded according to an

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[9] Post Operative Heat Stroke, by Moschowitz. (*Surgery, Gynecology and Obstetrics*, October, 1916.)

[10] Determination of the Comfort Zone, by F. C. Houghten and C. P. Yagloglou. (ASHVE TRANSACTIONS, Vol. 29, 1923, p. 361.)

[11] Comfort Standards for Summer Air Conditioning, by F. C. Houghten and Carl Gutberlet. (ASHVE TRANSACTIONS, Vol. 42, 1936, p. 215.)

[12] Cooling Requirements for Summer Comfort Air Conditioning, by F. C. Houghten, F. E. Giesecke, C. Tasker and Carl Gutberlet. (ASHVE TRANSACTIONS, Vol. 43, 1937, p. 145.)

[13] Summer Cooling Requirements of 275 Workers in an Air Conditioned Office, by A. B. Newton, F. C. Houghten, Carl Gutberlet and R. W. Qualley. (ASHVE TRANSACTIONS, Vol. 44, 1938, p. 337.)

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arbitrary index previously used by the Laboratory, in which ideal comfort is given an index rating of "4," feelings of slightly cool and slightly warm "3" and "5," respectively, and decidedly cool and decidedly warm for comfort "2" and "6," respectively. The additional two indices used in some of the earlier studies, namely, "1" for decidedly cold and "7" for decidedly hot, were not included, since it was deemed undesirable and also unnecessary to subject attendants or patients to these extremes. In all, 348 opinions of

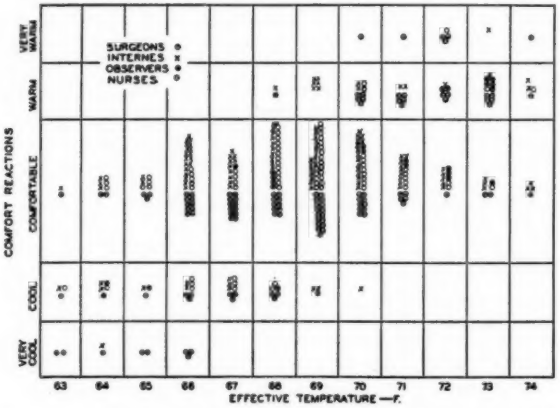

FIG. 2. RELATION BETWEEN THE EFFECTIVE TEMPERATURE OF THE AIR CONDITION MAINTAINED IN THE OPERATING ROOM AND THE NUMBER OF SURGEONS, INTERNES, OBSERVERS AND NURSES FEELING COMFORTABLE, SLIGHTLY COOL, SLIGHTLY WARM, VERY COOL AND VERY WARM

feelings of warmth were obtained from doctors, nurses and internes during the summer test period, and a somewhat lesser number of observations concerning the absence or degree of sensible perspiration. All of these individual observations are plotted against the effective temperature of the atmospheric condition in Fig. 2.

This gives a plot characteristic quite similar to identical plots obtained from earlier Laboratory studies for normal occupants of an assembly room. The great preponderance of number of observations of comfort appears in the center portion of the chart with decreasing numbers of observations to the right and left. Feelings of slightly cool "3" and slightly warm "5" are found, respectively, more or less to the left and to the right of the center, while still farther to the left and to the right are scattered observations of too cool "2" and too warm "6." From this chart a temperature range somewhere around 67 to 70 deg ET would readily be selected as including the most desirable conditions. However, the fewer observations obtained at higher and lower temperatures somewhat mask the general effect of the concentration of the data.

Separate symbols are used in Fig. 2 to indicate doctors, internes and nurses. While no doubt there is some difference in the temperature at which each group is comfortable, there were not sufficient individual observations made during the brief study to conclusively demonstrate this difference. However, the segregation of the data for the three groups tends to show that surgeons and internes both indicated maximum comfort at about 68 deg ET, while the nurses indicated maximum comfort at a little above 69 deg ET.

Fig. 3 gives a better numerical picture of the true relationship between the comfort indices and the effective temperature of the atmospheric condition. In this figure the percentage of observations expressed by the 5 indices, rather

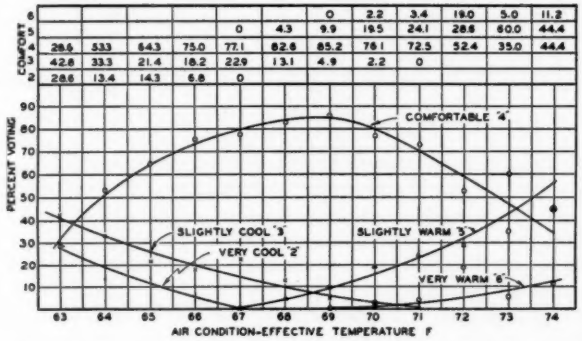

FIG. 3. RELATION BETWEEN THE EFFECTIVE TEMPERATURE OF THE AIR CONDITION MAINTAINED IN THE OPERATING ROOM AND THE PERCENTAGE OF ITS OCCUPANTS VOTING COMFORTABLE, SLIGHTLY COOL, SLIGHTLY WARM, VERY COOL AND VERY WARM

than their actual number, are plotted against effective temperature. Index "4," or ideal comfort, gives a parabolic type of curve, showing a minimum number feeling *ideal comfort* at about 62 deg ET. This curve then rises to a maximum of about 85 per cent feeling *ideal comfort* between the effective temperatures of 68 and 69 deg. With additional rise in effective temperature the curve falls until only 35 per cent are comfortable at an effective temperature of 74 deg. This curve is characteristic of all those obtained by the Laboratory in similar earlier studies. The fact that the curve is more steep on the cool side than on the warm side indicates that surgeons and attendants in an operating room, like persons in any type of audience hall, more readily sense discomfort due to a given number of degrees rise in temperature from that giving optimum comfort. The curve is characteristic also in that 100 per cent comfort is never found for any given condition; even when more than 80 per cent of all individuals indicated a feeling of *ideal comfort*, there were always some who felt either too warm or too cool.

The slightly cool "3" and decidedly cool "2," and the slightly warm "5" and decidedly warm "6" percentage curves are also characteristic. The slightly

cool and slightly warm rise more rapidly on either side of the condition for ideal comfort; while the decidedly cool and decidedly warm rise first slowly and then more rapidly, with wider variations in air conditions on the cool and warm side.

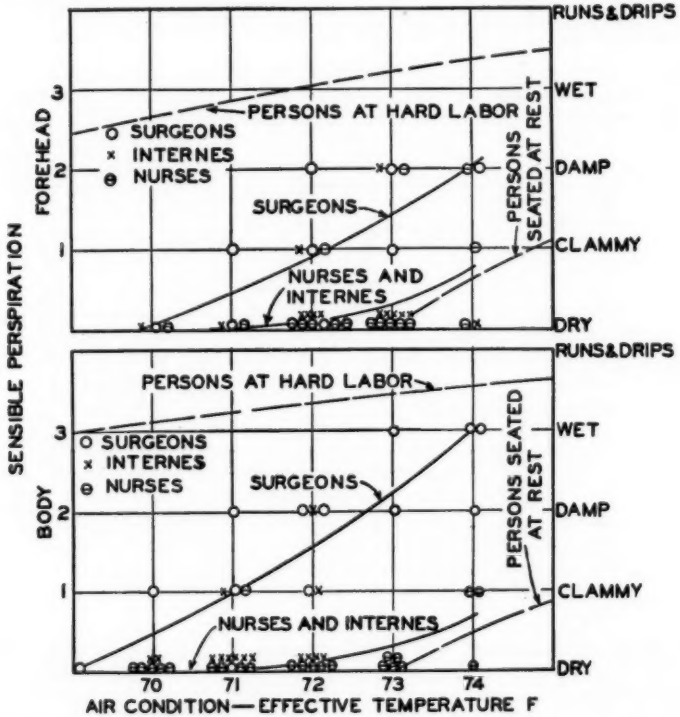

FIG. 4. SENSIBLE PERSPIRATION ON BODY AND FOREHEAD OF SURGEONS, INTERNES AND NURSES WHILE PARTICIPATING IN OPERATIONS WITH AIR CONDITIONS IN THE OPERATING ROOM AT INDICATED EFFECTIVE TEMPERATURES AND RELATIVE HUMIDITY OF APPROXIMATELY 50 PER CENT.

Broken line curves from other laboratory studies for persons seated at rest and persons at hard labor in atmospheres at the indicated effective temperatures and a wide range of relative humidity shown for comparison.

The parabolic curve of percentages indicating comfort in Fig. 3 is somewhat broader than those found in most other Laboratory studies for seated occupants in places of assemblage. This is probably due to the somewhat varying intensity of activity of those in attendance at operations. Undoubtedly in a major operation the surgeon and his more responsible attendants exert a

greater energy, resulting in comfort and higher rates of metabolism at lower temperatures. This accounts not only for the somewhat greater spread of curve but also in part for the slightly lower effective temperature than usually found at which the maximum percentage of workers feels comfortable. The increase in the clothing worn, particularly the masks covering parts of the head and face, and radiant heat resulting from intense illumination, and the close proximity of the workers to each other should also serve to account for comfort at the lower temperature. The point between 68 and 69 deg ET at which the doctors and their attendants were more often comfortable is significantly lower than the 70 to 73 deg ET, with an average of about 72 deg, that was obtained for seated occupants in a summer cooled room as found in earlier Laboratory studies. The slightly higher temperature at which the nurses were comfortable is in agreement with earlier findings which show that for persons seated at rest in a comfortable room, women desire a temperature of from 1 to 2 deg higher than men.

The sensible perspiration curves in Fig. 4 are drawn for both body and forehead perspiration of the surgeon, internes, and nurses, while participating in operations with air conditions in the operating room at the indicated effective temperatures and relative humidities in the neighborhood of 50 per cent. An arbitrary index for sensible perspiration, as used in previous studies at the Laboratory, was used in this present hospital study. An index rating of "0" is given to a dry condition of the skin, "1" for clammy, "2" for damp, "3" for wet, and "4" for the condition when the perspiration runs and drips down the surface of the skin. As mentioned previously, it was thought undesirable and also unnecessary to subject the patients and their attendants to extreme atmospheric conditions; hence, no votes were obtained for the "4" perspiration index.

As would be expected, the curves for the surgeons for both body and forehead perspiration are higher than the corresponding ones for the nurses and internes, this being due most likely to the greater activity and mental strain of the surgeons during the performance of the operation. Also, agreement with former studies is noted in the fact that the curves for body perspiration are located slightly higher than the corresponding curves for forehead perspiration.

For comparison, the broken line curves from former laboratory studies for persons seated at rest and persons at hard labor are superimposed on Fig. 4. Whereas the curves for the present hospital study were obtained in tests made at the indicated effective temperatures with a relative humidity of approximately 50 per cent maintained throughout, the curves from the former studies are the averages for a wider range of relative humidities. Nevertheless, this figure serves quite well to indicate the relative position of the hospital curves in comparison with the former studies. It must be kept in mind, however, that the hospital curves, especially those for the surgeons, are located by means of a relatively few points. Further study will serve to locate these curves more accurately.

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ACKNOWLEDGMENT

The authors highly appreciate the valued cooperation of Miss J. J. Turnbull, Superintendent of the Magee Hospital, Dr. W. S. McEllroy, Acting Dean of the School of Medicine of the University of Pittsburgh, and Dr. T. Lyle Hazlett, Head of the Department of Industrial Hygiene of the School of Medicine of the University of Pittsburgh, their cooperation having been necessary for the success of the study. Appreciation is extended also to the Magee Hospital Board, who made the necessary space available and assumed a considerable portion of the cost of remodeling the rooms and installing the air conditioning equipment.

The authors acknowledge their indebtedness to Doctors C. J. Barone, B. Z. Cashman, M. B. Ferderber, A. W. Sherrill, and J. W. Shirer, as well as the rest of the surgical staff, for their valued medical advice and suggestions, and to Prof. C. M. Humphreys, G. S. McEllroy and H. W. Reding, who aided in the design and installation of the experimental air conditioning system.

Acknowledgment is given also to the kind cooperation of the hospital management, including the engineering staff, for maintaining the conditions required for the tests, and the surgeons and their attendant staffs using the operating room for their cooperation, at times under trying circumstances, in aiding in the collection of the data on which this paper is based.

DISCUSSION

DR. B. Z. CASHMAN:[14] From the viewpoint of the surgeon, the original purpose of air conditioning the operating room and recovery ward in this hospital was to keep the patient in an atmosphere of relatively optimum constant temperature and humidity with frequent changes of air during the operation and the first few days of recovery, in contrast to conditions on the outside, which are subject to extreme and sudden changes. We were also concerned about the dangers of anesthetic explosions, and in hot weather the very real danger of perspiration dripping from the surgeon and assistants, with the possibility of it soiling the wound or the operative field.

There is no doubt that the patients have been more comfortable, and it is particularly noticeable to them in hot weather, when they are removed back to non-conditioned wards. A reasonable degree of warmth is desirable for the patient, but excessive heat and perspiration are exhausting to both the paitent and the surgeon. One's adaptability to abrupt changes or to extremes in temperature and humidity is impaired during illness, anesthesia, and loss of vitality as in old age, malnutrition, or following operation or injury, so that a constant suitable temperature and humidity is very desirable, at least until a certain degree of recovery has taken place.

Recovery from an operation, prolonged and severe, or in a handicapped patient, is uncomfortable and dangerous enough at the best, without having the patient subjected in addition to an unfavorable and uncomfortable environment, and especially extremes of heat which involve discomfort, exhaustion and undue loss of fluid at a time when the patient is unable to take the usual amount of fluid by mouth, and when the patient's resistance is lowered. There was noted less restlessness, less perspiring and less efforts to throw off bed clothing, particularly in hot weather. It was the ob-

[14] School of Medicine, University of Pittsburgh, Pittsburgh, Pa.

servation of the superintendent of the hospital that this was the quietest post-operative ward that she had ever known.

It has been a great comfort to the surgeon to be free of the dangers of dripping perspiration. Various methods are used by surgeons to minimize this danger, such as postponing elective operations during the heat spells, or the wearing of anything from gauze bands around the forehead to coils for circulating cool water, but the most prevalent and simple method is the delegation of a nurse to the duty of mopping the surgeon's face. This is by no means 100 per cent effective, so that it is a source of worry throughout the operation after the dripping stage has been reached. In our present system of air conditioning we work as comfortably in summer as in winter and the mop brigade has been abolished.

It also makes the surgeon's work much less exhausting than wroking in overheated, stagnant air, filled with anesthetic vapor, conditions which are not limited to summer but may be present at any time of year.

It is our belief and observation that regardless of what can be shown or proved physiologically, the air conditioning of the operating room and recovery ward, within the limits of temperature, humidity and air change, in which we have been working, has given comfort, a sense of well-being and a lessening of exhaustion, which improves the efficiency of the surgeons and aids in the recovery of the patient.

DR. M. B. FERDERBER: I think one point that was brought out some years ago when spinal anesthesias were first attempted, was that the number of post operative pneumonias would be decreased. A rather amazing analysis of this in 1932 by Dr. Arthur Dean Bevan, Rush Medical School, revealed the fact that post operative pneumonias were not decreased due to the use of spinal anesthesia. Irrespective of the failure to find definite physiological changes, but yet taking the observations and the word of a careful, conservative surgeon as Dr. Cashman, I believe the biggest virtue, besides the operating room, is the recovery room. Instead of moving a patient from one temperature of an operating room to a hall which has another temperature, through a corridor or hall into the main hall, to a drafty elevator and then through another hall which has an entirely different temperature into a room which may be different from any of the foregoing, it appears that the compatible conditions of both rooms might obviate post operative pneumonias.

When you know that post operative pneumonias instead of being purely a matter of inhalation of anesthesia from the gases may be caused by other factors, I am sure that a careful resumé will find there are just as many from spinals. Therefore I believe the greatest virtue in our venture in Pittsburgh, and considerably as important as the operating room, is the recovery room to eliminate the passage of the patient to varying conditions in the elimination of post-operative pneumonia.

DR. R. W. KEETON: [15] I should like to express my interest in this report. I expect to see more of these air conditioned operating and recovery rooms over the country. First of all, I consider it important that the patients do not suffer ill effects. It has been taught that an anesthetic disturbs the heat regulating mechanism, and would necessitate certain precautions to prevent a fall in body temperature. It is worthwhile knowing that this does not occur in the anesthetized subject when he is draped in the usual manner.

I believe that the results obtained are better than the statistical analysis indicate. With a study of a larger number of patients analyses will be more favorable.

We have been impressed by putting an occasional patient in our air conditioned room, with certain conditions that respond very definitely to air conditioning. The patient with hyperthyroidism often is forced to eliminate 100 per cent more heat

[15] Head, Dept. of Medicine, College of Medicine, University of Illinois, Chicago, Ill.

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than the normal subject. It is obvious that he should react favorably to an atmosphere in which heat is lost easily. This doubtless accounts for the fact reported that none of the patients with hyperthyroidism developed *thyroid crises* postoperatively.

If a group of patients with border line heart disease were operated in a controlled atmosphere and a similar group in a hot humid one, I am sure that differences would be apparent. The normal heart and circulation has such a great reserve capacity that adjustments can be made to unfavorable conditions. The crippled heart will not be able to make these.

Patients with asthma, and edematous states within the lungs do well in a cool relatively dry atmosphere. The moisture in the lungs prevents the easy passage of oxygen across the alveolar walls. Hence the individual with the water logged lung suffers from a lack of oxygen. The cool dry atmosphere apparently reduces the turgescence of the alveolar walls and allows the subject to again acquire oxygen in adequate quantities.

We are now asking the Univeristy to air condition a small ward for purposes of long time observations on patients suffering from various types of disease.

DR. CHARLES SHEARD:[16] I should like to express my appreciation of this excellent paper. No subject has been presented and discussed in this session which is of greater importance than the air conditioning requirements of an operating room and of the rooms into which patients are placed after operation. For, after all is said and done, anything and everything that tends to make surgery safer and post-operative recovery more certain is worthy of attention, time and repeated investigation. When any of us come into the hands of the surgeon and have to pass under his knife, we do so in the hope that we may be 100 per cent successful in every sense of the word success, for we want to continue to be with him and our families and friends. So anything that contributes, in a direct way or as an accessory factor, to aid the surgeon and his assitsants in their work, and anything that may furnish more favorable circumstances for an *uneventful recovery* are certainly worthy of much investigation.

Drs. Keeton and Cashman have pointed out very clearly some of the considerations which are of primary importance. As Dr. Cashman said, the experimental set-up which has been reported has made it possible for the surgeon to get rid of sweat and mops, so that, with less distraction and interruption he can carry on and need have regard only for the mops and sponges that must be used when any of us have to go to surgery.

The question arises as to what shall be considered fairly representative and desirable conditions of temperature and humidity. One of my colleagues and I have been interested for some time in attempting to answer the question and have obtained data which we hope to supplement and present to the Society at a later date. We know that, under ether vasoconstriction of the peripheral vascular vessels is as fully released as is possible, or, in other words, the peripheral blood vessels are open and vasodilated. One of the simplest methods of finding the degree of vasodilatation in the extremities, as evidenced by skin temperatures measured with thermocouples, is to use ether anesthesia. After about 20 min. under ether, the peripheral blood vessels will be dilated to the maximal amount. The heat-regulating mechanisms are thrown out of commission and, therefore, there are no controls to prevent the loss of heat from the body. On the other hand, there is usually a rise in the rectal temperature; at any rate, a decrease in body temperature is not common. In the past, I judge that surgeons have been somewhat uncertain as to what environmental

[16] Director, Division of Physics and Biophysical Research, The Mayo Foundation and The Mayo Clinic, Rochester, Minn.

conditions were most favorable to the patient, since decreases in body temperature and excessive loss of heat from the body might be considered as detrimental. To be sure, the surgeon has at hand blankets or covering which he may use as he sees fit. In general, a normally moist, red, warm skin, especially of the extremities, is the criterion.

It is my belief that a dry-bulb temperature of 80 F to 82 F and a relative humidity of about 40 per cent are ideal air conditioning standards for both operating rooms and wards. Such conditions may be maintained with but little change throughout the year. I think that they are satisfactory and comfortable working conditions for the surgeon, and are a fair compromise from the standpoint of the patient who has nothing much to do for a few days except to live in anticipation of getting well and being discharged. Furthermore, the patient may be restricted in diet and exercise may be at a minimum. For these and other reasons I believe that he will be comfortable and under satisfactory environmental conditions, from the standpoint of a non-working individual, if the temperature is 82 F (or slightly higher) with a relative humidity of about 40 per cent.

I wish to say again that the report of these investigators is an excellent introduction to a very important series of researches which should be carried out in various localities. I am sure that we are all appreciative of the many hours of data gathering that these men have spent. In general, surgeons are, very properly, not receptive of added personnel of a useless character insofar as the operating room procedures are concerned, and do not welcome added contraptions devised by the inquisitive minds of research workers. However, I am sure that surgeons and experimenters must have a very harmonious and glorious time together, for they have a worthwhile undertaking in common.

COL. W. A. DANIELSON: From the standpoint of design the matter of air circulation becomes very important. I wonder if any thought was given to that question. Also the location of the ultraviolet radiation lamps is important. Dr. Hart at Duke University has the lamps placed so they shine directly on the patients while performing the operation. In that way he has been able to perform pleural cavity operations without any seroius results, or with much less serious results than previously. The lamps may be placed in the ducts, or directly above the patients.

F. C. HOUGHTEN: The paper was well presented and adequately discussed and there is little that I can add. Referring to questions by Colonel Danielson concerning the rate of air change and the ultraviolet radiation lamps, it should be understood that this was distinctly an experimental system. It was designed to be flexible, in order to lend itself to experimental work. While the system was designed to give a wide range of air change, the rate of air supply used in most of the study gave about six changes an hour.

As we have studied hospital and particularly operating room practice, we find these conditions exist; very frequently there are many uncontrolled sources of heat and therefore even in the winter time occupants suffer from too high air temperatures. Then, again, in order to decrease the spread of infection by drafts of air stirring up dust, and bacteria, the operating room is closed up tightly giving the very minimum possible air change. As a result of this, one of the important factors that Dr. Cook mentioned occurs; the concentration of anesthetic throughout the room builds up to the point where the doctor as well as the patient is gassed. With air conditioning including air cleaning it is found possible not only to get normal air circulation and air change as considered necessary in a room of this kind but an even higher air change so as to cut down the anesthetic concentration throughout the room.

The ultraviolet radiation lamp was directly over the operating table and is similar to the one used at Duke and mentioned by Colonel Danielson.

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We, as engineers, highly appreciate the cooperation that we have had from the medical profession as indicated by the several papers presented at this meeting, and we are particularly pleased in Pittsburgh with the close cooperation given on the part of the doctors.

I am sure John Howatt, who a few years ago was instrumental in initiating the first cooperation between the Society and the medical group, is highly pleased with the course that this cooperation has taken during the past two years. We can accept it as a very good indication of what we may hope for in the near future. Certainly during the past year the degree of cooperation and good will between the medical people and our research has increased several hundred per cent.

E. H. DAFTER: I believe Colonel Danielson was probably interested in whether or not recirculation was permitted, or whether all outside air had to be used, when considered from the sanitary viewpoint. Were any experiments carried on with this in mind?

MR. HOUGHTEN: Particularly in the operating room 100 per cent outside air was supplied. This air was first filtered, with a better than ordinary dry filter, then passed through an electric precipitator, and then through the washer, or the humidity control unit.

In the recovery ward they used up to 50 per cent recirculation, but here again the amount of outside air was much greater than would have been the case in any ordinarily ventilated ward.

No. 1112

SOME REFLECTION AND RADIATION CHARACTERISTICS OF ALUMINUM

By CYRIL S. TAYLOR* AND JUNIUS D. EDWARDS** (*MEMBER*)
NEW KENSINGTON, PA.

ALUMINUM has proved to be an excellent reflecting material for radiation in all parts of the spectrum. Its high reflectivity for both visible and ultra-violet radiation, combined with its non-tarnishing characteristics, have won its adoption for coating reflectors for telescopes and other optical instruments. The development of the aluminum reflector,[1] which is given both brightness and permanence by a duplex electrolytic treatment, has afforded aluminum wide application in the lighting field. The high reflectivity and correspondingly low emissivity of aluminum for infra-red radiation have been responsible for the application of aluminum foil in insulating structures. While considerable data have been accumulated as to the performance of aluminum in the visible range, there is less known about its behavior with infra-red radiation, and particularly the effect of surface films upon its reflectivity and emissivity.

The measurement of radiation from surfaces at relatively low temperatures (approximately 100 F) presents a number of difficulties. Several satisfactory emissivity meters have been described in technical literature,[2, 3] but apparatus for such measurements is not generally available from instrument makers. In this paper there is given a description of an instrument suitable for making measurements of the emissivity of surfaces at temperatures up to 300 F or higher, together with a variety of measurements on aluminum obtained by the use of this instrument. The general plan of this instrument follows closely that of an emissivity meter designed by R. E. Gould, to whom acknowledgment

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* Physical Chemist, Aluminum Research Laboratories, Aluminum Company of America.
** Asst. Director of Research, Aluminum Research Laboratories, Aluminum Company of America.

[1] Aluminum for Reflectors, by J. D. Edwards. (*Transactions, Illuminating Engineering Society, 29*, p. 351, 1934.)

[2] An Apparatus for the Measurement of the Total Normal Thermal Emissivity of Sheet Materials in the Temperature Range 140 to 500 F, by Paul F. McDermott. (*Review of Scientific Instruments, 8*, 1937, p. 185.)

[3] Measurement of Radiation from Aluminum Surfaces, by E. Schmidt. (*Hauszeit. V.A.W. Erftwerk A.G. Aluminium, 2*, 1930, p. 91.)

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is made. Advantage was taken of a number of details of design described by other workers, and no special originality is claimed for the equipment.

DESCRIPTION OF EMISSIVITY METER

The emissivity meter comprises a heater which heats the sample being measured to a definite uniform temperature, a radiometer which receives energy radiated from the specimen, and a potentiometer system by which the electro-

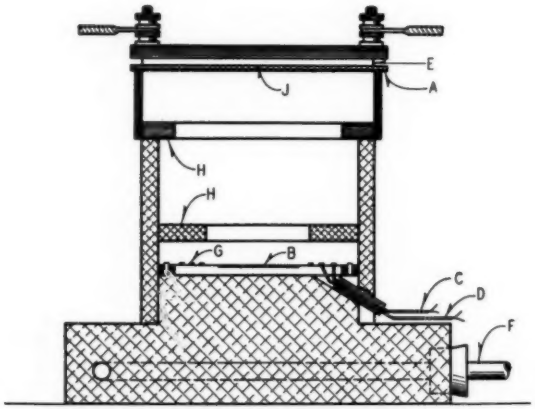

FIG. 1. SECTIONAL VIEW OF EMISSIVITY METER:

A. Sample for emissivity measurement
B. Thermopile receiving radiation from *A*
C. Thermocouple leads measuring temperature of thermopile cold junctions
D. Thermopile leads
E. Electric heater plate for maintaining sample at uniform temperature
F. Inlet for constant temperature water supply circulating in base
G. Copper pins to which are attached cold junctions of thermopile
H. Radiation shields with aluminum surface
J. Thermocouple junction for measuring sample temperature

motive forces generated by the radiometer and various essential thermocouples are quantitatively measured. The general construction and operation of the meter can best be described by reference to Fig. 1, which shows a cross-section of the instrument.

The specimen holder and heater unit (*E*) is placed on top facing downwards, so as to eliminate convection currents as far as possible. The heater face-plate is a piece of aluminum sheet, oxide-coated. It is heated by another sheet, oxide-coated for insulation, wound with nichrome ribbon and backed by a similar-sized insulating block, pressed against it and bolted together. A copper-

constantan thermocouple, embedded in a small groove cut in the face of the heater in such a way that the junction (J) is located at the center of the face, permits the temperature of the hot face to be measured. The heater, which is usually operated at a temperature of 100 F, requires about 4w (watts) power. The sample to be measured (A) is fastened to the face of the heater by means of a thin coat of cement.

The radiometer consists of a copper-constantan thermopile (B) unit of 18 junctions mounted on the central portion of a water-cooled (F) aluminum block. The central cylindrical part of the black is 3$\frac{1}{16}$ in. in diameter and projects $\frac{3}{4}$ in. above the rest of the block, which acts as a base. A tight-fitting, thick-walled aluminum tube is forced onto the projecting cylinder. This tube extends upwards, 2 in. above the thermopile, and has fastened in it, $\frac{3}{8}$ in. above the thermopile, a radiation shield (H) consisting of an aluminum ring $\frac{1}{4}$ in. thick and with a circular opening 1$\frac{5}{8}$ in. in diameter. On top of the aluminum tube is fastened another radiation shield (H) made of a synthetic resin ring, $\frac{1}{4}$ in. thick and with an opening having a diameter of 2$\frac{1}{2}$ in. The exposed bottom of this ring is coated with aluminum foil to reduce as much as possible any radiation from this source. A thin tube of synthetic resin, approximately the diameter of the aluminum tube, is mounted on top of the ring to act as a low-heat-conducting support for the heater.

The receiver of the thermopile is a lacquered disk of aluminum foil, 1$\frac{1}{4}$ in. in diameter and 0.0003 in. in thickness, to which are cemented 18 thin copper segments, each of which has soldered to it a thermocouple of No. 40 B. & S. gage copper-constantan wire. The copper segments were cut from a copper disk 0.0003 in. thick and they covered practically the whole area of the receiver. The assembled receiver disk is blackened with smoke from burning camphor. The thermopile leads are shown at (D). The cold junctions are soldered to heavy copper pins (G) inserted in a synthetic resin ring. A copper-constantan thermocouple, soldered to one of the cold junction pins, allows the temperature of the cold junction to be obtained (C). The whole thermopile unit was built and assembled, following very closely the directions given by Heilman.[4]

In calibrating the emissivity meter, a piece of bright aluminum foil was fastened to the heater plate to establish the radiometer reading with a surface of low emissivity. A sheet of black paper was employed to establish the radiometer reading for a point of high emissivity. The aluminum foil was assumed to have an emissivity of 3 per cent at 100 F, and a value of 95 per cent was assigned to the black paper. The radiometer readings for surfaces of intermediate emissivity were established by exposing various proportions of foil and black paper as the radiating surface. This was accomplished by cementing to the aluminum foil, pie-shaped sectors of black paper.[5] These were always attached as diametrically opposite pairs of equal size, so as to avoid any possible error resulting from lack of exact centering. In this way a surface could readily be obtained whose average emissivity was any value between 3 and 95 per cent.

In this work the standard of reference for low emissivity was bright aluminum foil with an assigned value of 3 per cent at 100 F. In this paper

[4] Surface Heat Transmission, by R. H. Heilman. (*Mechanical Engineering, 51*, 1929, p. 355.)
[5] A Reflectometer for All Types of Reflecting Surfaces, by Frank Benford. (*General Electric Review, 37*, 1934, p. 457.)

the emissivity values are expressed as a percentage of the emissivity of a *black body* at the same temperature. There is very good evidence in the measurements of McDermott,[6] H. Schmidt and Furthmann,[7] Coblentz[8] and E. Schmidt[9] that the emissivity is as low as 3 per cent and possibly lower. If another value is eventually established for the emissivity of aluminum foil, the emissivity values reported in this paper can be corrected accordingly. The commercial aluminum foil with an emissivity of 3 per cent has a purity of

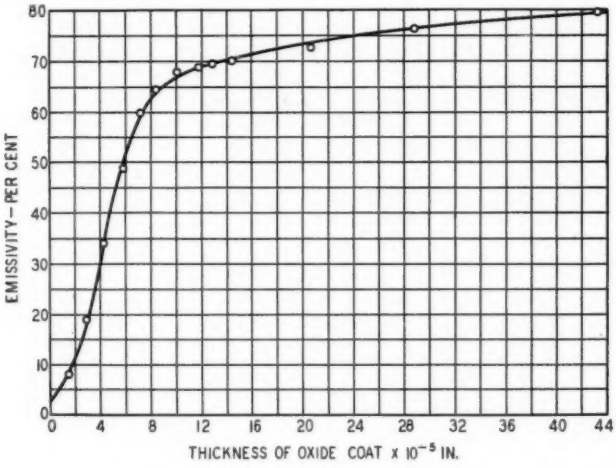

FIG. 2. RELATION BETWEEN THICKNESS AND EMISSIVITY (AT 100 F) OF AN ELECTROLYTICALLY PRODUCED OXIDE COAT ON ALUMINUM

about 99.4 per cent. Schmidt's data at various temperatures showed an increase in emissivity of one sample of aluminum from 4.0 per cent at 200 C to about 5.0 per cent at 500 C. Recent measurements from this laboratory gave a value of 2.6 per cent for the emissivity of aluminum, 99.996 per cent pure.[9a]

EFFECT OF OXIDE FILMS UPON EMISSIVITY

In general, non-metallic substances exhibit high emissivity properties, and aluminum oxide is no exception to this general rule. It might be expected, therefore, that the oxidation of aluminum surfaces would increase their emis-

[6] Loc. Cit. See Note 2.
[7] Radiation from Solid Bodies, by H. Schmidt and Furthmann. (Mitt Kaiser-Wilhelm Inst: Eisenforsch. Abhandl., *10*, 1928, p. 225.)
[8] Radiometric Investigations of Infra-red Absorption and Reflection Spectra, by W. W. Coblentz. (*Bureau of Standards Bulletin 2*, 1906, p. 457.)
[9] Loc. Cit. See Note 3.
[9a] The Properties of High Purity Aluminum, by C. S. Taylor, L. A. Willey, D. W. Smith and J. D. Edwards. (*Metals and Alloys*, *9*, 1938, p. 189.)

sivities. With normal atmospheric oxidation, however, the films are usually so thin as to have little effect upon emissivity. The oxide film, moreover, is always present and all emissivity measurements on aluminum are for a surface covered with an oxide film, even though it be extremely thin. Various estimates of the thickness of the oxide film naturally present upon aluminum are of the order of 1 to 2 $\times$ 10^{-6} cm (approximately 0.4 to 0.8 $\times$ 10^{-6} in.).[10, 11]

The oxide film upon aluminum can be increased in thickness and changed in character by various chemical and electrochemical treatments. The thin dielectric films produced upon aluminum by anodic oxidation, for use in electrolytic condensers, are frequently of appropriate thickness to show interference colors. Substantially thicker films are applied for decorative and protective purposes, and these films in general range from about 0.0001 in. to 0.0006 in. (0.0025 mm to 0.015 mm) in thickness. In Fig. 2 is shown the relationship between emissivity and thickness of film for a series of oxide films electrolytically produced in a 15 per cent sulfuric acid electrolyte at 70 F.

It will be noted from the curve that the emissivity increases linearly with increase in thickness of the oxide coating up to about 8 $\times$ 10^{-5} in. Above about 10 $\times$ 10^{-5} in., the emissivity shows only a slow change with increase in thickness; the emissivity has already reached a high value at this point. All oxide coatings do not show the same emissivity values at the same thickness; the emissivity may be greater or smaller than the values of Fig. 2, depending upon the structural characteristics of the coating, which are controlled by the electrolyte employed and the coating conditions. These differences may be substantial.

The thickness of the oxide coatings shown in Fig. 2 was measured in several ways. When the coating had a thickness of about 10 $\times$ 10^{-5} in. or greater, it was easy to measure the thickness directly by observing a cross-section of the foil and coating under the microscope. The technique of preparing the sample for examination is described in an earlier report.[12] Some very thin oxide coatings showed interference colors. In these cases it was possible to measure their thickness by the method of Blodgett and Langmuir,[13] using as color standards a series of built-up films of known layers of barium stearate on polished chromium. Dr. K. B. Blodgett kindly made such a comparison and estimated the thickness of the oxide films on two samples of aluminum foil, which had been electrolytically oxidized in boric acid electrolyte, to be 19 and 26 $\times$ 10^{-6} cm, respectively. The thickness was also evaluated by measuring the increase in weight resulting from oxidation and calculating the thickness from the measured area and the density of the oxide. This method gave thickness values of 19 and 27 $\times$ 10^{-6} cm, which are quite comparable with those measured by Dr. Blodgett on similar samples.

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[10] Some Optical Observations on the Protective Films on Aluminium in Nitric, Chromic and Sulphuric Acids, by L. Tronstad and T. Höverstad. (*Transactions, Faraday Society, 30*, 1934, p. 362.)

[11] Second Experimental Report, Section V, Aluminium, by W. H. J. Vernon. (*Transactions, Faraday Society, 23*, 1927, p. 152.)

[12] Methods of Testing Oxide Coatings on Aluminum, by J. D. Edwards. (*Proceedings, American Society for Testing Materials, 37*, Appendix II, Report of Committee B-7, 1937, p. 261.)

[13] Built-up Films of Barium Stearate and Their Optical Properties, by Katherine B. Blodgett and Irving Langmuir. (*Physical Review, 51*, 1937, p. 964.)

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According to the Wien displacement law, the energy peak for surfaces radiating at temperatures of about 100 F occurs at a wave-length of about 9.3μ (0.009 mm). Apparently, therefore, the oxide-coated surface only behaves like the massive oxide when the coating thickness reaches a value approaching the same magnitude as that of the wave-length of the radiation being reflected or emitted.

THE EMISSIVITY OF LACQUER-COATED ALUMINUM

In Fig. 3 is given a series of data showing the emissivity of aluminum foil coated with lacquer films of various thicknesses and colors. The lacquer in all of these films was formulated with a certain pyroxylin base to which had been added a small amount of resin. For the colored films, various dyes were added to give scarlet, yellow, green and blue films. The presence of the dye seemed to have little effect upon the emissivity as compared with the clear coating, and there was no specific effect due to any of the particular colors. By comparing the emissivity values of Fig. 3 with Fig. 2, it will be observed that the lacquer coatings in general have less effect in increasing the emissivity than does an oxide coating of similar thickness.

Other lacquer films were observed in which the color was obtained by incorporation of a pigment. In general, the aluminum foil bearing these films showed emissivities intermediate between the foils with the dyed lacquer films and the oxide films. The thicknesses of the lacquer films were measured by

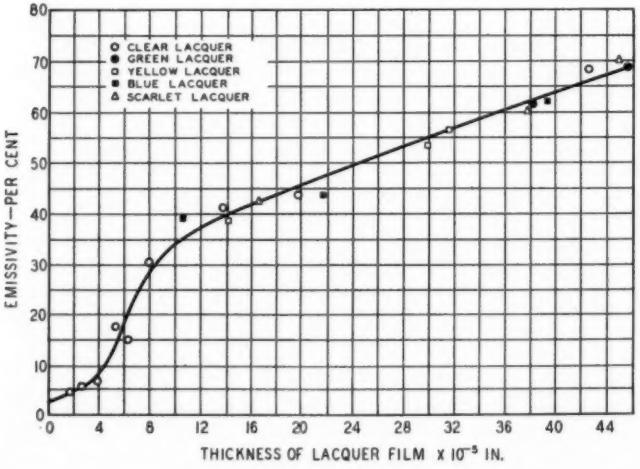

FIG. 3. RELATION BETWEEN THICKNESS AND EMISSIVITY (AT 100 F) OF LACQUER FILMS, CLEAR AND COLORED WITH DYES, ON ALUMINUM

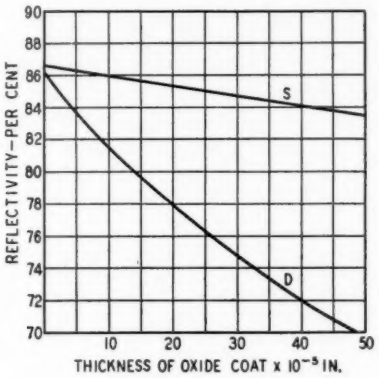

FIG. 4. REFLECTIVITY FOR LIGHT OF ALUMINUM SURFACES WITH ELECTROLYTICALLY PRODUCED OXIDE FILMS OF DIFFERENT THICKNESSES; THE *S* GRAPH IS FOR SPECULAR REFLECTORS AND THE *D* GRAPH IS FOR DIFFUSE REFLECTORS

determining the increase in weight of the coated foil, caused by the presence of the lacquer, and calculating the thickness from the density of the dried lacquer.

It is quite practical to coat foil commercially with a uniform, thin film of lacquer which does not raise the emissivity above a value of 10 per cent. Such a coating of proper composition offers substantial protection against deterioration where the service is such that prolonged contact with moisture is anticipated.

REFLECTIVITY FOR VISIBLE LIGHT

While the eye may be a good judge of the reflectivity of a surface for light, it cannot always appraise the value of a surface for reflecting infra-red radiation. This can be demonstrated by a few selected examples. In the case of the data on oxide films shown in Fig. 2, the emissivity increases from a value of 3 per cent to over 75 per cent; that is to say, the reflectivity for infra-red radiation falls from a value of 97 per cent to less than 25 per cent. For an opaque body, the sum of the reflectivity and emissivity at the same wave-length is 100 per cent. While a substantial change in emissivity has taken place with increases in thickness of the oxide coating up to about 30×10^{-5} in. the changes in reflectivity for visible light have been of minor character. Fig. 4 shows the effect on reflectivity for light, of increase in thickness of the oxide coating for two types of oxide-coated surfaces: one of these coatings is

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specular in character and the other diffuse, and the metal in each case is of different purity. However, even in the case of the coating showing the greatest change, the loss in reflectivity for a coating thickness of 30 x 10^{-5} in. was only from 86 to 75 per cent.

The reflectivity for visible light was measured by the method of A. H. Taylor,[14] which gives the total reflectivity directly without the use of any standard reflecting surface. The equipment employed was the instrument designed by Baumgartner,[15] which employs a light sensitive cell for measur-

TABLE 1—COMPARISON OF EFFECT OF COLORED LACQUER ON ALUMINUM FOIL UPON REFLECTIVITY FOR VISIBLE AND INFRA-RED RADIATION

DESCRIPTION OF SAMPLES	THICKNESS OF COATING, INCHES × 10^{-5}	REFLECTIVITY FOR LIGHT, PER CENT	REFLECTIVITY FOR INFRA-RED RADIATION[a], PER CENT
Lacquer, clear	20	80	57
Lacquer, clear	43	82	33
Lacquer, with black pigment	30	5	30
Lacquer, with black pigment	68	5	8
Lacquer, with white pigment	40	67	31
Lacquer, with white pigment	93	76	12
Lacquer, with yellow dye	20	76	43
Lacquer, with yellow dye	50	68	21
Lacquer, with blue dye	30	8	45
Lacquer, with blue dye	64	5	22
Lacquer, with red dye	27	28	39
Lacquer, with red dye	63	23	26

[a] Radiating surface at 100 F.

ing the brightness of the sphere wall. Commercial aluminum foil with an emissivity of 3 per cent has a reflectivity of about 85 per cent for visible light.

The data of Fig. 3 show that while the thickness of a lacquer coating has an important effect upon the emissivity of the surface, the presence of a dye in the lacquer has no appreciable effect upon emissivity. The presence of the dye may, however, have a very profound effect upon reflectivity for light; data on this point are shown in Table 1. These figures show that a surface may have low reflectivity for light and fairly good reflectivity for infra-red radiation, or high reflectivity for light and poor reflectivity for infra-red radiation.

The lacquer films, measurements of which are shown in Table 1, are intensely colored. It is quite possible to coat with thinner lacquer films, and aluminum

[14] A Simple Portable Instrument for the Absolute Measurement of Reflection and Transmission Factors, by A. H. Taylor. (*Bureau of Standards Bulletin*, Bureau of Standards Scientific Paper No. 405, 1920.)

[15] A Light-sensitive Cell Reflectometer, by G. R. Baumgartner. (*General Electric Review, 40*, 1937, p. 525.)

with these lacquer films may have high reflectivity (above 80 per cent) for infra-red radiation and still be decidedly colored because of the intense coloring action of the dye. Comparing the lacquers with black and white pigment, it will be noted that samples, showing as widely different reflectivities for light as 5 per cent and 67 per cent, show substantially the same reflectivity for infra-red radiation.

EFFECT OF SURFACE ROUGHENING

The assumption is commonly made that a highly polished metal surface is necessary for low emissivity. This assumption has, as a background, the classical theory which calls for an increase in emissivity as a surface is roughened.[16] Experimental measurements, however, demonstrate that roughening of a very pronounced degree may take place without any very substantial increase in the emissivity value. In Table 2 are given a series of emissivity values for foil which has been roughened in various ways. The samples include foil which had been embossed with various patterns for decorative effect, and also foil which had been roughened with an abrasive cloth (No. 120 Aloxite cloth). While it is true that the abrasive did not make the foil very rough, nevertheless its specular reflecting characteristics were completely destroyed. Among all of these samples, the highest emissivity measured was 6.4 per cent. Tests were also made on foil which had been roughened by etching in sodium hydroxide solution for different times and finish-dipped in nitric acid, according to a procedure commonly employed in the aluminum industry. Six samples roughened in this way all had diffusing surfaces, and only one sample showed an emissivity as high as 5 per cent. All of these samples were so roughened as to destroy the polish and specular reflecting characteristics. Of course extreme methods of roughening might be employed which would give substantial increases in emissivity, but surfaces of that character are not ordinarily found.

TABLE 2—EFFECT OF SURFACE ROUGHENING UPON EMISSIVITY

DESCRIPTION OF SAMPLE	EMISSIVITY AT 100 F, PER CENT
Embossed foil—Pattern No. 1	4.5
Embossed foil—Pattern No. 2	5.6
Embossed foil—Pattern No. 3	4.7
Embossed foil—Pattern No. 4	5.6
Embossed foil—Pattern No. 5	6.1
Bright foil roughened with abrasive cloth	4.4
Bright foil roughened with abrasive cloth	6.3
Bright foil roughened with abrasive cloth	6.6
Bright foil etched in hot sodium hydroxide solution for times of 0.5 to 2 min. and dipped in nitric acid	3.5 4.0 5.0

[16] Heat Radiation of Technical Surfaces at Ordinary Temperatures, by E. Schmidt. (*Gesundheit.-Ing. Beihefte Series 1*, No. 20, 1927.)

EFFECT OF EXPOSURE UPON EMISSIVITY

The very penetrating character of infra-red radiation was illustrated in the preceding section by the measurements on emissivity and reflectivity of oxidized aluminum surfaces. In quite another field, the penetrating characteristics of infra-red radiation are illustrated by the aerial photographs which have been taken by Captain Albert Stevens. With infra-red sensitized photographic plates, views of mountains as far as 331 miles distant from the photographer have been taken, though the mountain itself was invisible to the eye because of the intervening haze and fog.

Various surface films may be formed upon aluminum as the result of exposure to corrosive attack. Fortunately, the oxide film formed on aluminum in the

TABLE 3—EMISSIVITY OF ALUMINUM FOIL WITH DULLED SURFACES

SAMPLE NO.	DESCRIPTION OF SAMPLE	EMISSIVITY PER CENT RADIATING SURFACE AT 100 F
1	Aluminum foil after 2 years' exposure to salt spray and moisture in boat-house on the seashore	10
2	Aluminum foil-covered cardboard taken from foil-insulated dry-ice cabinet; exposed to weather on beach at Coney Island from September, 1934, to April, 1935	4
3	Aluminum foil, chemically oxidized by treating with hot solution of sodium carbonate and chromate	9
4	Aluminum foil with thin lacquer coat heated to partially decompose lacquer and color it brown	10
5	Aluminum foil suspended vertically in laboratory for 3 years and measured with accumulated dust and fume	5

atmosphere is quite impervious, and this limits its growth and thickness. This oxidation may therefore occur without an appreciable effect upon the reflectivity of the surface for infra-red radiation. The eye, of course, is no judge as to the infra-red radiation characteristics of a surface and surfaces, which to the eye may be dull and poorly reflecting, may still be good reflectors for radiation of long wave-length.

As a quantitative measure of the effective difference between the reflection factor of aluminum for visible light and infra-red radiation, the following example may be cited: A sample of aluminum foil was exposed in the laboratory near a window for three years. This sample was allowed to accumulate dust and fume for that period, and its reflectivity for visible light was then measured without any preliminary cleaning. It was found after the three-year period of exposure to have lost 15 per cent in reflectivity for visible light; its infra-red reflection factor, however, had not changed appreciably. Origi-

nally, it was about 97 per cent, and with the three years' accumulation of dust and fume, its reflectivity had only been reduced to 95 per cent.

In June, 1938, Prof. Gordon B. Wilkes of Massachusetts Institute of Technology removed samples of aluminum foil insulation from the underside of a roof of a home in Wellesley Hills, Mass., where the insulation had been installed in November, 1933. Samples from each of the four layers of crumpled foil appeared to be in perfect condition and untarnished. The average emissivity of the four samples of foil which had been in service for five years was 5.4 per cent as compared with a value of 4.5 per cent which Professor Wilkes used for the new foil.

A series of emissivity measurements of aluminum, which had been exposed to the weather or chemically oxidized and treated in various ways, is given in Table 3.

SUMMARY

1. A simple instrument for measuring emissivity is described and a variety of measurements presented to show the effect of surface films on the emissivity of aluminum at 100 F.

2. The oxide coating naturally present on aluminum is too thin to have an appreciable effect upon emissivity. Thick oxide coatings can be produced electrolytically, however, which substantially increase the emissivity. The relation between oxide thickness and emissivity for one type of coating is shown by a graph.

3. Thin lacquer films can be applied to aluminum, which are protective in nature but not thick enough to raise the emissivity above a value of about 10 per cent. The relation between thickness and emissivity is presented in a graph.

4. Colored dyes in lacquer films have little effect upon emissivity, but pigments have an appreciable effect.

5. The eye may be a good judge in appraising the reflectivity of a surface for visible light but cannot evaluate the infra-red radiation characteristics. A piece of aluminum may have high reflectivity for visible light and low reflectivity for infra-red radiation, or it may have only fair reflectivity for light and be an excellent reflector of infra-red radiation, depending on the presence or absence of surface films. It may also be a good reflector for both kinds of radiation.

6. Surface roughening does not necessarily produce a surface of high emissivity and a bright polished aluminum surface is not an inherent requisite for low emissivity.

7. Because the natural oxide film on aluminum is quite impervious and very slow to grow in thickness, substantial weathering of aluminum foil can occur with little increase in emissivity.

8. Samples of aluminum foil used for thermal insulation under various conditions of service have shown very little change in emissivity over long periods.

DISCUSSION

PROF. E. A. ALLCUT (WRITTEN): This paper is of particular interest in view of the many applications of reflective surfaces for heat insulation purposes, and one very important aspect of it is the possibility of the emissivity of such surfaces increasing with time, corrosion, and with the deposition of moisture and dirt. These questions are referred to in the last section of the paper, and in this connection an

experiment performed at the University of Toronto may be of interest. It was made in a guarded hot plate, 24 in. square. Two similar air cells were placed on either side of the plate and each was divided by a vertical partition down the center, parallel to the hot plate. These partitions consisted of corrugated paper with thin aluminium foil on one side and plain brown paper on the other, and the following tests were made:

1. The heat counductance was measured with the foil facing the hot plate and a conductance of 0.188 Btu per square foot per degree Fahrenheit temperature difference per hour was obtained.

2. This test was then repeated with the partitions reversed so that the foil faced the cold plates, all other conditions being the same, when a conductance of 0.22 Btu was obtained.

3. The foil was then covered thickly with dust from a domestic vacuum cleaner, left for 15 hours and then raised gently to the vertical position for testing. The film of dust remaining on the surface was so thin that it was practically invisible when viewed at right angles to the surface. It was visible when viewed at an angle of about 45 deg to the surface. Test 1 was then repeated, whereupon the measured conductance was 0.254 Btu, being an increase of 13½ per cent above that of the clean surface.

It is recognised that the heat transmission in this case was partly by radiation and partly by convection, and also that single tests are not always to be relied on, but the tests described above were essentially a single run made in three parts, with the same apparatus, under exactly the same test conditions, so that the above comparison appears to be a fair one.

P. D. CLOSE (WRITTEN): I note that the experiments were conducted at a temperature of 100 F. It would appear therefore that the results would be directly applicable for only a small percentage of the time during the summer months. Is there any evidence to prove whether or not these results would be applicable to average winter temperature conditions of 35 or 40 F. The transfer of radiant energy is of course a function of the fourth powers of the absolute temperatures of the emitting and receiving surfaces.

P. F. MCDERMOTT[17] (WRITTEN): The authors are to be congratulated on the excellent paper, both to content and type, especially because basic papers dealing with fundamentals which are not generally understood are entirely too rare.

There still is much which might be told about metallic, reflective type, insulations to remove the mystery and magic which tends to surround them. To fully acquaint engineers and subsequently laymen with these materials, their correct value and proper use, quite a number of papers might still be required. Several points come to mind which seem to be worth noting at this time, however.

We have three criticisms of the method of emissivity measurements:

1. Radiation Shields in view of the specimen will alter the energy impinging on the receiver if the shields are not at the temperature of the receiver. This is not a constant factor but a function of the reflectivity of the specimen which tends to make all surfaces yield the same value as measured. We believe the upper shield of resin to be subject to criticism in this respect.

2. Calibration was done by an ingenious method of considerable merit. It appears, however, that a surface of known value might have been used in place of the black paper which was assumed to have an emissivity of 95 per cent.

[17] Physicist, Johns Manville Research Laboratories, Manville, N. J.

3. Had the authors included emissivity data on wood, asbestos paper, and other materials, we would be in better position to evaluate the data reported.

Films, Lacquers and Color: We are of the opinion that the evidence of somewhat different emissivities existing for different films, lacquers and colors is due to two factors: (1) Orientation of the particles of solids, and (2) The effective thickness of the film. These characteristics of any film would be difficult to ascertain. For example the effective thickness we think of might be quite different than that obtained from assuming uniform coverage and calculating from the film weight per unit area and from its density. Similarly some arrangement of particles such as in metallic blacks might account for the variation.

We know, however, that except for bright metals all materials in sufficient thickness to be opaque are very good radiators and that, therefore, films of any of them influence the emission of a poor emitter very much and more or less equally so. In other words, oxides, lacquers, organic pigments, dust, grease and so on, all are good emitters and if present in an effective thickness in excess of 4 to 5 μ will cause the surface to be a good radiator also, regardless of which one of the materials is present. For example, a scarcely visible film of lacquer applied over clean bright metal by an air brush will generally produce a surface having an emissivity of over 25 per cent and a second coat, an emissivity near 80 per cent.

Color we know to be unimportant, white and black paints are usually nearly the same; occasionally the white being the blacker.

It is in this respect that so called reflective insulations defy visual evaluation either for the purpose of control in production or that of judging by consumers. As indicated by Messrs. Taylor and Edwards, films on metal can be of such nature and thickness to destroy any insulating value derived from a low emissivity, with no visible indication of the loss. Comparable losses in the efficiency of other insulations can scarcely be obtained without essentially complete destruction of the material and relatively small variations can be detected by simple visual and manual inspection.

Corrosion: The authors have discussed oxide films of various thicknesses but have assumed throughout that no pinholes resulted from the oxidation or corrosion. Their statements are true and no further discussion would be in place if metals were used in substantial thicknesses. However, foil corrosion frequently results in pinholes through the foil. In this case the holes increase the apparent emissivity to a much greater extent than their area might indicate. In fact, since the cavity behind the holes is usually at quite a different temperature than the foil, the open areas may be several times more effective than equal areas of a black body. Thus 10 per cent of open area might result in an apparent emissivity of perhaps 50 per cent.

Unprotected metallic foils of a few ten thousandths inch in thickness are known to fail in this way, often in a few months' exposure to normal atmospheres such as might exist within insulated walls. Counts of as high as 600 holes per square inch have been made with apparent emissivities well over 50 per cent.

Protective films retard the corrosion insofar as they are continuous and do not lose their value due to oxidation, flaking and chalking. Long practice in applying coatings to metal have familiarized engineers with the difficulty in obtaining permanent adhesion and complete permanent coverage.

Roughening of Surface: As pointed out by the authors, metallic surfaces may appear quite rough and still have low emissivity. Again we are confronted with a physical property impossible of evaluation except by some laboratory measurement. We would like to report to the authors that the emissivity of foil can be materially increased by crumpling and opening it out again for measurement.

Visible Light: Much of our discussion has been directed to the fact that visible light measurements either by the eye or otherwise cannot be relied upon to indicate

a surface's optical properties with respect to infra-red. A point not mentioned is that sometimes we have need for a material of certain characteristics with respect to each of these two wave-length regions.

For example there are occasions when a surface should be used which has a high reflectivity for visible or relatively short infra-red and high emissivity for longer infra-red, or normal heat radiation. For example, the coolest roof is obtained with an outer surface of white paint or asbestos which absorb about an equal quantity of the incident solar energy as metal but which radiate far more back to the sky. This is a point very worthwhile considering in the treatment of exterior surfaces of buildings, tanks, etc., which are subject to intense sunshine. The subject was discussed some years ago by Dr. W. W. Coblentz of the U. S. Bureau of Standards.

PROF. F. B. ROWLEY: Many questions have been raised as to the proper emissivity values to use in practice for bright metallic surfaces. The authors are to be complemented in presenting some practical data in this field.

The fact that the values are all relative and based on aluminum with an emissivity coefficient of 0.03 would probably make but little difference in the practical application of the values.

In Fig. 2 the emissivity coefficients are given for various thicknesses of oxide, and in Fig. 3 they are given for various thicknesses of lacquer film. While the thicknesses of these coatings are given it is not apparent as to how they would be correlated to practical conditions. Would it be practical to put on three or four coats of lacquer or would that give too thick a film? How long a time would be required under ordinary conditions to get a thickness of oxide as represented by 8 on the horizontal scale of Fig. 2?

It would be very interesting and helpful if the authors would use the same test apparatus and get emissivity coefficients for other materials used in practice.

COL. W. A. DANIELSON: Two questions of life of the bright foil. One exposed to the weather, as are the oil tanks in the South, and the other in a wall space. I am wondering if the writers have any information based on experience.

Another point is the question of painting radiators with aluminum paint. There is a common impression that you cut down emissivity quite materially but I understand there is a way of keeping the aluminum appearance without loss of emissivity.

PROFESSOR ROWLEY: Several years ago we ran a series of experiments with radiators painted with different types of paint and placed in a vacuum tank. Aluminum paint was included in this series. We found that aluminum paint cut down the total amount of heat given off by radiators by about 12 per cent and that this reduction was entirely in the amount of radiant heat given off. That is, the aluminum paint had no appreciable effect on the amount of heat given off by convection.

G. W. PENNEY: In Fig. 3 of the paper, the emissivity of aluminum covered with lacquer films is shown down to a lacquer thickness of 2×10^{-5} in. No points are shown for colored lacquer film below 10×10^{-5} in. thickness. Can colored films be produced down to 4×10^{-5} in.? If so, will the colored films give the same reflectivity as the clear lacquer, *i.e.*, emissivities below 10 per cent?

J. D. EDWARDS: In answer to Mr. Close's question, I would call attention to the fact that the emissivity coefficients presented in this paper are relative values expressed as a percentage of the emissivity of a black body and they are, therefore, applicable to both summer and winter temperatures. The paper does give some published values of emissivity at 200 C and 500 C as compared with 100 F, and these values show that the temperature coefficient is quite small.

The experiments described by Professor Allcut are very interesting. The question might be raised, however, as to whether a thick layer of dust taken from a vacuum cleaner reproduces a surface condition which would be representative of aluminum foil in actual service. The data of Professor Wilkes on aluminum foil taken from actual installations suggests that the dust which collects under these conditions does not affect the insulation value appreciably.

Mr. McDermott's remarks are interesting as he is an authority on the measurement of emissivity and the value we selected for the emissivity of the bright aluminum foil was largely based on measurements which he has published.

Mr. McDermott's criticism of our use of an upper radiation shield made of synthetic resin does not apply because, as pointed out in the paper, the exposed bottom surfaces of the shield were covered with aluminum foil to reduce to a minimum any radiation from this source.

Both Mr. McDermott and Professor Rowley spoke of emissivity measurements on other materials. No doubt in time we will be able to contribute something along these lines, although for the present most of our work has been done on aluminum foil.

Mr. McDermott also referred to the fact that a coat of clear lacquer, which changed the appearance very little, increased the emissivity to a value over 25 per cent. That may be true, yet it has been shown that clear lacquer applied to aluminum foil in production consistently produced a product which had an emissivity of not over 10 per cent. The laboratory controlled that characteristic by emissivity measurements made every day on representative samples. The thickness of coating, whether it be lacquer or other material, is the important consideration. The greatest care was taken in making the thickness measurements and they were checked in a variety of ways. No difficulty is experienced in securing good adhesion and covering with thin lacquer films. Practical experience shows that pinholes in aluminum foil for insulation are not much of a problem.

Mr. McDermott's statement that reflective insulation "defies visual evaluation either for the purpose of control in production or that of judging by consumers" seems to go too far. If a lacquer coating is being applied, some method of measurement other than the eye must be employed to control its thickness. Such control is, however, very simple and precise in the factory where the application is made. However, most aluminum foil is used without a lacquer coating, and both the producer and the consumer can rest assured that as long as the surface is bright there has been no appreciable change in insulating effectiveness. Furthermore, the data which we have presented in this paper show that even substantial oxidation and discoloration can take place in the surface appearance without any appreciable loss in infra-red reflection efficiency. Other types of insulation are not immune to change in insulating value as a result of settling, moisture absorption, etc.

Professor Rowley raises the question of how long it would take to get an oxide coating sufficiently thick to have an emissivity of approximately 65 per cent. Under ordinary service conditions it would be impractical to get such a thickness by natural oxidation. The lacquer films described in Fig. 3 were applied by a roller coating machine and the thickness of film was controlled by the adjustment of the machine.

Colonel Danielson has asked a question as to the relative life of foil on oil tanks and in the wall space of a house. We have reason to believe that, under ordinary conditions, aluminum foil would last as long as the house. On oil tanks, where the foil is subject to constant attack by the weather, it is an open question as to what you would predict as the normal life.

The question of painting radiators with aluminum paint is one which comes up from time to time. As Professor Rowley has explained, there is a reduction in the rate of heat dissipation of something on the order of 10 to 12 per cent. That, however,

is not a loss of heat. Some of the modern thinking on radiators looks towards a construction which increases the amount of convection from the radiator and decreases the amount of heat dissipated by radiation. The heat which is radiated by a radiator to an outer wall is largely lost, as is heat which is radiated to the ceiling and to the floor. A great many modern radiator installations provide a closure or a shield over the radiator which cuts off most of the radiation to the room. A coat of clear lacquer over the aluminum paint will raise its emissivity without appreciable change in the appearance.

Mr. Penney asked a question regarding colored lacquer films as thin as 4×10^{-5} in. The dyes used for coloring lacquers have such high tinctorial power that you can get a light-colored film of the thickness mentioned.

No. 1113

A STUDY OF THE HEAT REQUIREMENTS OF A SINGLE-GLAZED TEST HOUSE AND A DOUBLE-GLAZED TEST HOUSE

By M. L. Carr,* R. A. Miller,** Leighton Orr,*** and David Shore,† *(MEMBERS)*

THIS paper reports the results of a study of the relative heat requirements necessary to maintain a temperature of 70 F in each of two similarly constructed test houses, one having single- and the other double-glazed windows. The investigation was planned and carried out by the Pittsburgh Plate Glass Co. with the aid and counsel of Pittsburgh Testing Laboratory during the period January 18 to April 22, 1933. The data were received for analysis in 1937 and were prepared for publication by the ASHVE Technical Advisory Committee on Air Conditioning Requirements of Glass, consisting of the following members: M. L. Carr, Chairman, R. A. Miller, Vice-Chairman, F. L. Bishop, A. N. Finn, S. O. Hall, E. H. Hobbie, F. W. Parkinson, Harold Perrine, W. C. Randall, L. T. Sherwood, J. T. Staples, C. Tasker, G. B. Watkins and F. C. Weinert.

Test Houses and Apparatus

The test houses were erected on the top of a three-story laboratory building at Creighton, Pa. The houses with relation to their surroundings are shown in Fig. 1. The exterior appearance of one of the houses is shown in Fig. 2, while Fig. 3 shows the interior of one of the houses.

The interior dimensions of each house were: length and width, each 10 ft, 4¼ in.; height, 8 ft, 6 in. The walls of the houses extended up from the roof of the building to which they were anchored, the floor of each test house being 2 ft above the roof and level with the top of the building parapet walls. The houses were frame structures, sheathed and stuccoed on the outside, with 1 in. insulating board and gypsum plaster on the inside. One-fifth pitch gable roofs with cypress shingles laid on building paper were used in the construction.

* Director, Pittsburgh Testing Laboratory, Pittsburgh, Pa.
** Technical Sales Engr., Pittsburgh Plate Glass Co., Pittsburgh, Pa.
*** Research Engr., Pittsburgh Plate Glass Co., Creighton, Pa.
† Junior Research Engr., ASHVE Research Laboratory, Pittsburgh, Pa.

Presented at the 45th Annual Meeting of the American Society of Heating and Ventilating Engineers, Pittsburgh, Pa., January, 1939.

FIG. 1. TEST HOUSES ON TOP OF THREE-STORY LABORATORY BUILDING WITH RELATION TO THEIR SURROUNDINGS

Each house contained seven double-hung windows, each window containing two 26-in. by 30-in. lights of double-strength glass, and one wooden entrance door. The southwest, northwest, and northeast walls each contained two windows, while the southeast wall contained one window and the door. The spacing of the lights in the double-glazed windows was ⅜ in., and venting of the air between the space and the outside was provided. The vents were closed during these tests. The floor of each house was painted gray, while the walls and ceiling were painted white.

The percentage of glass area to other areas for the two houses were as follows:

Glass area to total heat transfer area	13.3 per cent
Glass area to total side wall area	21.5 per cent
Glass area to floor area	70.6 per cent

HEATING EQUIPMENT

Each house was provided with a five-kilowatt electric unit heater as shown at the top of Fig. 3. The heating element of each unit was divided into two parts of three and two kilowatts, each of these having independent thermostatic control. The fan of each unit ran continuously, while the heating elements operated intermittently as the occasion demanded, being thermostatically controlled to hold the room at 70 F. The thermostats were located 42 in. above the floor in the center of the room and were protected from the sun's rays by a metal shield. The position of each unit was adjusted to

give substantially the same rate of air movement over each window. The average velocity of the air over the inner faces of the lights was between 1.5 and 2 mph, as measured by a Kata thermometer.

PROCEDURE

No attempt was made to build the houses unusually tight to avoid infiltration, but they were made as nearly equal in this respect as possible, and means were provided for checking this condition at intervals during the test. The method used was to apply a suction to each house and equalize the amounts of air drawn from each by sealing cracks. This was done at the beginning of the test and at monthly intervals thereafter.

The electrical energy required to heat each house was measured with a recording watthour meter, which was read daily at 11:30 a.m. The time during which the two-kilowatt and three-kilowatt elements in the units of each house were in operation was registered by recording wattmeters.

The temperature of each test house at a point near the thermostat and four feet above the floor was recorded. The average difference in temperature between the air at the center of the room and the inside surface of the window lights of each house was given by the thermocouples attached to the center of one of the lower window lights on each side of the house. These four thermocouples were connected in series, with their cold junctions in a reservoir

FIG. 2. EXTERIOR OF SINGLE-GLAZED TEST HOUSE. DOUBLE-GLAZED HOUSE IDENTICAL

at the center of the room. The thermocouple junctions were held against the glass surfaces with small drops of specially prepared cement.

Outdoor temperatures were recorded with a resistance thermometer, sheltered from the sun by a wooden housing, open only on the northeast side.

The wind velocity was measured midway between the test houses with a rotating cup anemometer coupled to an electric circuit and to a demand meter,

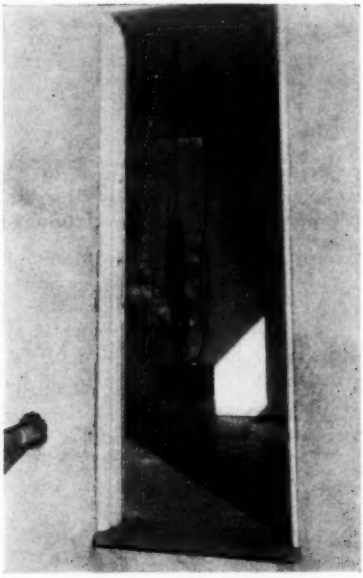

FIG. 3. INTERIOR OF SINGLE-GLAZED TEST HOUSE. INTERIOR OF DOUBLE-GLAZED TEST HOUSE IDENTICAL

utilized to record the number of revolutions of the cups in continuous five-minute periods. The device was calibrated with a rotating vane portable anemometer.

Sunshine intensity was recorded by means of a photronic cell connected to a recording millivoltmeter. The cell was attached to a disc driven by an electric motor, which caused it to follow the sun across the sky. The cell and motor were mounted outdoors under a protecting glass cover with the face of the cell normal to the sun's rays at noon. Provision was made for adjusting the inclination of the cell from time to time as the sun became higher in the sky with the advance of the seasons.

TEST RESULTS

The data obtained during the continuous 95-day test period were summarized by determining the average for each 24-hour period, beginning at 11:30 a.m., of the following:

(1) The daily average outdoor temperature was obtained by determining the average temperature for each hour from the recorded temperature chart, and averaging these 24 hourly averages.

(2) The temperature at the center of each house was taken from the recorder charts and did not vary much, being thermostatically controlled, except when the solar heat was more than sufficient to carry the heating loads.

(3) The daily average temperature of the inside glass surface for each house was computed, using a conversion from the millivolt recorder charts.

(4) The daily average wind velocity was obtained by dividing the total number of anemometer cup revolutions by the conversion factor.

(5) The sun intensity was determined from the area under the curve of the sunshine recorder by a planimeter, using the intensity at night as a base line. A square inch of sunshine area was arbitrarily taken as one unit of sunshine, and is so used in this report.

It should be emphasized that these values afford only a means of comparing one day with another with respect to sunshine, and do not give absolute values of solar radiation impinging on the various surfaces of the test houses.

A log of the daily average room temperatures, average inside glass surface temperatures, weather conditions and average daily electrical heat requirements of the test houses for the test period is plotted in Fig. 4. Also included in this plot are the calculated heat requirements of the two houses based on the accepted transmittances and film conductance coefficients of the materials used, and infiltration losses, assuming no solar heat input. The ratio of the electrical heat supplied to the houses also appears in this Figure. The average daily electrical input in Btu per hour is plotted against temperature difference between the inside and the outside in Fig. 5. The relationships between the average daily electrical heat supply in Btu per hour per degree Fahrenheit and the mean temperature, the wind velocity in miles per hour, and the sun intensity in units are plotted in Figs. 6, 7 and 8, respectively.

The relationship between the total daily heat supply in Btu per hour per degree Fahrenheit (electrical heat plus estimated solar heat) and the wind velocity in miles per hour is shown in Fig. 9.

The relationships of the ratios of the electrical heats supplied to the houses to the temperature difference between inside and outside, the wind velocity and the sun intensity, are given in Fig. 10.

DISCUSSION OF RESULTS

An examination of Fig. 4 will serve to substantiate the following generalities:

(1) The test houses were maintained at substantially 70 F at the center of each house except for a few times when this temperature was exceeded due to sun effect.

(2) The 24-hour average outdoor temperatures varied from 7 F to 65 F during the test period. The actual temperature extremes were 0 F and 82 F.

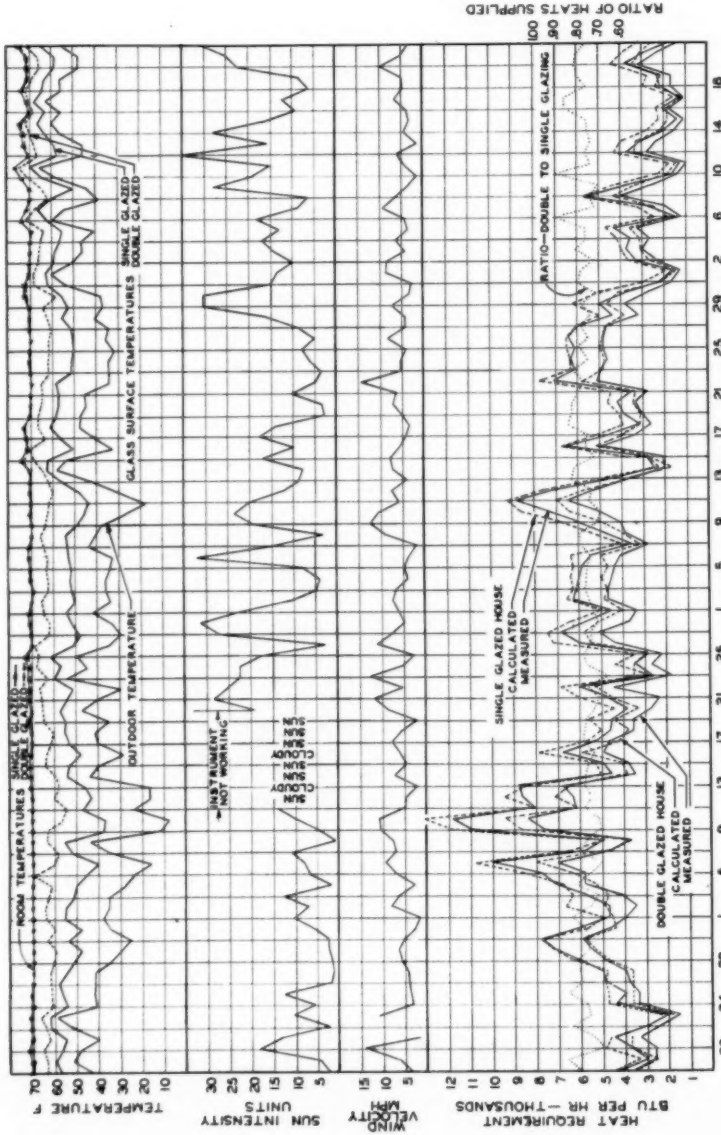

FIG. 4. LOG OF TEST DATA—JANUARY 18 TO APRIL 22, 1933

(3) The 24-hour average wind velocities varied from 1.5 mph to 14 mph. The extreme velocities were 0 mph and 32 mph.

(4) The average temperature of the inner surface of the glass in the single-glazed house remained about half-way between the inside and the outside temperatures, with little or no sunshine and wind, while the temperature of the inner surface of the

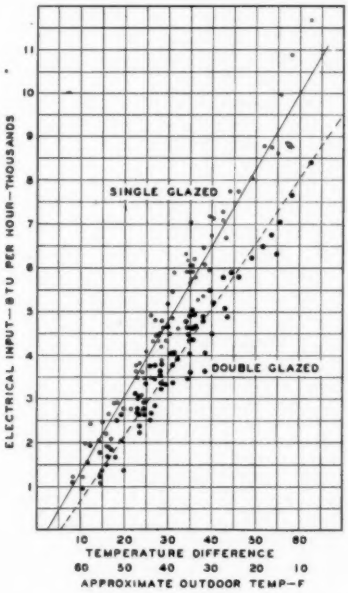

FIG. 5. RELATION BETWEEN MEAN TEMPERATURE AND ELECTRICAL INPUT IN BTU PER HOUR FOR SINGLE- AND DOUBLE-GLAZED HOUSES. APPROXIMATE OUTDOOR TEMPERATURE BASED ON AN ASSUMED INDOOR TEMPERATURE OF 70 F

inside lights of the double-glazed house was about one-quarter of the way between the inside and outside temperatures under the same conditions, being closer to room temperature.

COMPARISON OF MEASURED AND CALCULATED HEAT REQUIREMENTS

The daily variation of the measured electrical heat input in Btu per hour for both houses is shown in Fig. 4, together with the calculated heat requirements of the two houses based on the prevailing outside temperature and wind velocity and the accepted heat transfer constants taken from the ASHVE

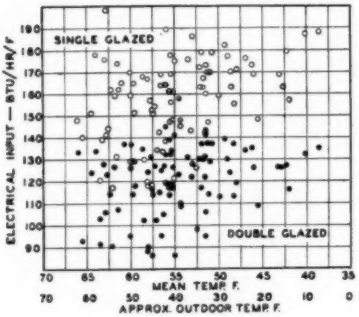

FIG. 6. RELATION BETWEEN MEAN TEMPERATURE AND ELECTRICAL INPUT IN BTU PER HOUR PER DEGREE FAHRENHEIT FOR SINGLE- AND DOUBLE-GLAZED HOUSES. APPROXIMATE OUTDOOR TEMPERATURE BASED ON AN ASSUMED INDOOR TEMPERATURE OF 70 F

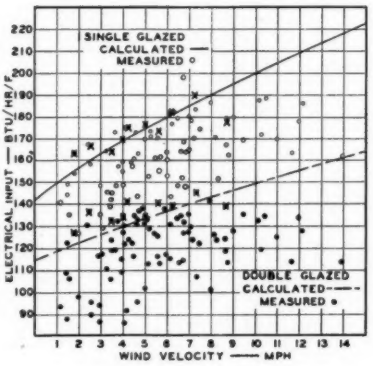

FIG. 7. RELATION BETWEEN WIND VELOCITY IN MILES PER HOUR AND ELECTRICAL INPUT IN BTU PER HOUR PER DEGREE FAHRENHEIT FOR SINGLE- AND DOUBLE-GLAZED HOUSES. CURVES GIVING CALCULATED REQUIREMENTS BASED ON PREVAILING TEMPERATURE DIFFERENCE AND WIND VELOCITY AND GUIDE TRANSMISSION COEFFICIENTS. NO ALLOWANCE FOR SUN RADIATION. *X*-POINTS REPRESENT ELECTRICAL INPUT ON DAYS WITH LITTLE OR NO SUNSHINE

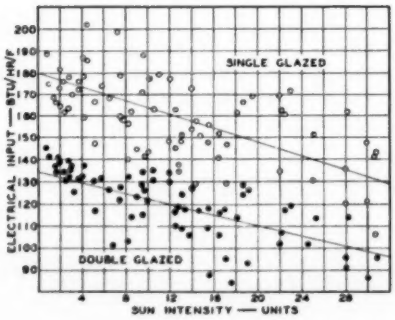

FIG. 8. RELATION BETWEEN SUN INTENSITY AND ELECTRICAL INPUT IN BTU PER HOUR PER DEGREE FAHRENHEIT. NO ALLOWANCE FOR WIND VELOCITY

HEATING, VENTILATING, AIR CONDITIONING GUIDE, no allowances being made for the sun effect. The effect of infiltration is also included in this calculation. The calculated curve is higher in most cases than the measured curve, as is to be expected, since the inclusion of solar heat in the calculated curve would lower its value. It should be noted that the differences between the two curves, generally speaking, are greater on days of more intense solar radiation.

VARIATION OF HEATING REQUIREMENTS WITH WEATHER CONDITIONS

The variation of the average daily electrical input in Btu per hour with temperature difference between the inside and outside for both houses is shown in Fig. 5. A lack of sufficient points in the low temperature difference range makes it rather difficult to predict this part of the curve. The two curves do not intersect the temperature difference axis at zero, indicating that sufficient amounts of heat were stored in the houses to keep them at 70 F when certain outdoor temperatures less than 70 F were attained. The curve for the single-glazed house intersects the temperature difference axis at about 2.5 F, while the one for the double-glazed house intersects it at about 5.5 F. This indicates that for these respective temperature differences no artificial heat was required to maintain the predetermined control temperature of 70 F. The fact that no heat was required for the double-glazed house at a temperature of about 64.5 F as compared with about 67.5 F for the single-glazed house indicates that the double-glazed house retained heat more effectively than did the single-glazed house.

The variation of electrical heat supplied to the houses in Btu per hour per degree Fahrenheit with mean temperature is shown in Fig. 6. For both the single- and double-glazed houses there was an apparent increase in the number of Btu per hour per degree Fahrenheit required with decreasing mean temperature, and therefore also with an increasing temperature difference between the inside and the outside. This is contrary to the accepted fact that the heat transfer through any material per degree temperature difference increases with increasing mean temperature. It is also contrary to the generally accepted belief based on a number of surveys that the heating requirements of a building per degree temperature difference between the inside and outside decreases with increasing temperature difference, or in other words that the efficiency of heat utilization increases with the heat demand. A possible explanation, at least for a part of this difference, is that in these tests there was a continuous thermostatically controlled electrical heat supply which operated only when the occasion demanded it. Thus, all of the heat supplied by the electrical heater was necessary to maintain the controlled temperature, and this required amount of heat per degree temperature difference decreased as the mean temperature approached that critical temperature at which no artificial heat was required. On the other hand, in the case of a home or an office building, where the occupancy requires heat only during certain hours of the day, a usual occurrence is the supply of more than sufficient heat, especially on warm days. This overheating raises the Btu per hour per degree Fahrenheit supplied above that necessary, causing a condition of less efficient heating at higher outdoor

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temperatures. The data plotted in Fig. 6 take no cognizance of sun radiation, wind velocity or infiltration. Infiltration may account for the indicated increase in heat demand per degree temperature difference for lower mean temperatures. This would not be the case if the volume of infiltering air were constant over the range of mean temperatures. However, since temperature difference or the *chimney effect* in a building serves as a motivating

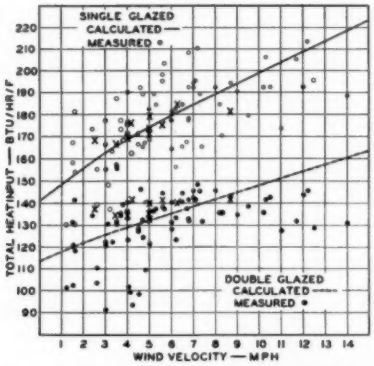

FIG. 9. RELATION BETWEEN WIND VELOCITY IN MILES PER HOUR AND TOTAL HEAT INPUT (ELECTRICAL + SOLAR HEAT) IN BTU PER HOUR PER DEGREE FAHRENHEIT FOR SINGLE- AND DOUBLE-GLAZED HOUSES. CURVES GIVING CALCULATED REQUIREMENTS BASED ON PREVAILING TEMPERATURE DIFFERENCE AND WIND VELOCITY AND GUIDE TRANSMISSION COEFFICIENTS, *X*-POINTS REPRESENT ELECTRICAL INPUT ON DAYS WITH LITTLE OR NO SUNSHINE

force to increase infiltration in addition to the wind effect, this may serve to produce the observed effect.

The variation of electrical heat supplied in Btu per hour per degree Fahrenheit with wind velocity in miles per hour is shown in Fig. 7. The points for both the single- and double-glazed houses indicate a definite increase in heat demand with increasing wind velocity. The calculated curves for the two houses are also shown, based on the assumed transmittance and surface conductance coefficients for the building materials used in the constructions, and coefficients of 0.88 to 1.42 and 0.51 to 0.66 for the single- and double-glazed windows, respectively, the variation being caused by a 0.15 mph range in wind velocity. These curves include heat input necessary due to infiltration losses. The scattering of points away from the calculated curve results from neglect of the solar radiation effect. Thus, the sunnier the day, the farther the heat input points are from the calculated curve. The points follow the general

directional trend of the calculated curve for velocities up to about eight miles per hour. For higher values of velocity, the points indicate a decreasing effect of velocity. This bears out earlier ASHVE Laboratory studies,[1] which indicate that the effect of wind velocity in increasing heat requirements of buildings may be over-emphasized in present calculation methods. The *X*-points are for days when there was little or no sunshine. For the single-glazed house these *X*-points conform quite well to the calculated curve; however, for the double-glazed house, with one exception, they lie above the calculated curve. This would indicate that the value of 0.51 to 0.66 used for the transmission coefficient in the double-glazed case may be too low.

The relationship between the electrical heat supplied in Btu per hour per degree Fahrenheit and sun intensity in units is given in Fig. 8. For both the single- and double-glazed houses there was a definite decrease in the amount of electrical heat required with increasing sun intensity. The scattering of the points was due, in part, to the wind velocity effect of which no account was taken in this plot.

An attempt was made to combine the wind velocity and sun intensity effects by adding to the measured heat supply an amount of heat proportional to the sun intensity for any test day. This procedure should tend to eliminate the sun effect for any given day, and thus give the value of heat required under a condition of no solar radiation. The result of this attempt is shown in Fig. 9 in which the measured electrical input plus 1.55 times the solar intensity for the single-glazed house and 1.10 times the solar intensity for the double-glazed house is plotted against the wind velocity. The *solar intensity multipliers* represent the negative of the slopes of average curves drawn through the points of Fig. 8. On this basis the points plot better, tending to arrange themselves about the calculated curves for the single- and double-glazed houses. These calculated curves are the same as those which appear in Fig. 7, while the *X*-points again indicate days when there was either little or no sunshine.

The variation in the ratio of the heat required for the double- and single-glazed houses with temperature difference between the inside and outside, the wind velocity and the sun intensity is shown in Fig. 10. Heat loss from either building should be proportional to the temperature difference multiplied by the integrated transmittance coefficients for the ceiling, the floor and the walls, including the windows. Since such integrated coefficients must vary with single and double glazing, the ratio between the heat losses by conduction through the structure should be constant for any temperature difference, and, therefore, other factors being equal, the ratio between the heat requirements of the double- and single-glazed houses should be constant, insofar as it is affected by transmittance through the walls. This, however, is not shown to be the case in Fig. 10 (top plot), which indicates a higher ratio for lower temperature differences. This variation may be due to the effect of wind and sun intensity. There is an apparent greater dispersion of the points upwards for low temperature differences, with a constant value of approximately 0.70

[1] Wind Velocities Near a Building and Their Effect on Heat Loss, by F. C. Houghten, J. L. Blackshaw and Carl Gutberlet. (ASHVE TRANSACTIONS, Vol. 40, 1934, p. 387.)

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for the lower limit. It may be that this lower ratio of 0.70 is indicative of the transmittance effect, and the upward dispersion of the points the effect of wind velocity and sun intensity.

Wind velocity should have two effects on the ratio between the heat required for the double- and single-glazed houses. Other things being equal, infiltration into the two houses should be independent of single or double glazing, and, in fact, in the two houses under consideration an attempt was made to make infiltration equal. Hence, an equal increase in heat loss by infiltration for both houses independent of glazing should tend to increase the ratio with increased

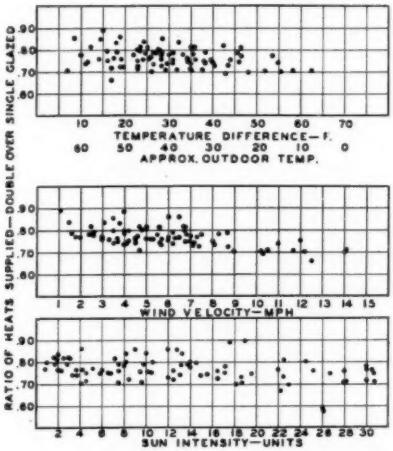
RATIO OF HEATS SUPPLIED—DOUBLE OVER SINGLE GLAZED
.90
.80
.70
.60
10 20 30 40 50 60 70
TEMPERATURE DIFFERENCE—F.
60 50 40 30 20 10 0
APPROX. OUTDOOR TEMP.
1 2 3 4 5 6 7 8 9 10 11 12 13 14 15
WIND VELOCITY—MPH
2 4 6 8 10 12 14 16 18 20 22 24 26 28 30
SUN INTENSITY—UNITS

FIG. 10. RELATION BETWEEN TEMPERATURE DIFFERENCE, WIND VELOCITY AND SUN INTENSITY AND THE RATIO OF THE HEATS SUPPLIED TO THE HOUSES (DOUBLE- OVER SINGLE-GLAZED)

wind velocity. Increase in wind velocity also increases transmission through windows by increasing the outside film conductance coefficient. Since the outside film conductance coefficient for the double-glazed window is a smaller proportion of the combined effect for the entire window than in the single-glazed case, the increased wind velocity is accompanied by lesser increase in transmission through the double-glazed than through the single-glazed window. This effect should decrease the ratio with increasing wind velocity. Evidently the latter effect was greater than the former since the ratio decreases with increasing wind velocity as shown by the points of Fig. 10 (middle plot).

The effect of sun radiation should be the same in reducing the heat requirements for the two houses, insofar as all forms of construction other than the glass are concerned. Either single or double glazing has a relatively small

effect in intercepting solar radiation through the windows. From the best data available,[2] each thickness of glass may be expected to intercept approximately 10 per cent of solar radiation. This means that approximately 90 per cent of the solar radiation incident against a window should pass through the single glazing, and 81 per cent through double glazing. Hence, it would be expected that the effect of sun radiation should be to increase slightly the ratio of the heats supplied. On the other hand, a double-glazed window is a better *heat trap* than a single-glazed window. Consequently, while slightly less heat can get in by direct radiation, considerably less can get out by transmission. The bottom plot of Fig. 10 shows a slight decrease rather than increase in the ratio of the heats supplied with increasing sun effect, indicating that the greater *trapping* effect of the double glazing overbalances its greater absorption.

CONCLUSIONS

1. By reason of the fact that under the same conditions the inside surface of the glass in the double-glazed house will be warmer than the same surface in the single-glazed house, occupants of a double-glazed house will feel more comfortable than those in a single-glazed house. This fact also means that frost will form more readily on the inside surface of the glass in the case of a single-glazed house than it will in the case of a double-glazed house.

2. The double-glazed test house described required from 70 to 80 per cent as much heat as the single-glazed test house. These ratios were affected somewhat by wind velocity, temperature difference and sun intensity, the ratio decreasing with increase in wind velocity and sun intensity, and with an increase in temperature difference.

3. The values shown for the ratio of the heats supplied to the test houses would apply only for the ratio of wall to window area and for the particular construction of these test houses. Had either this ratio or the construction been different, then the ratio of the heats supplied would also have been different. As the ratio of window area to wall area increases the ratio of the heats supplied to the two houses would decrease, the limit being the case of an all-glass house, in which case the ratio would be about 50 per cent. Likewise, if the ratio of window area to wall area is held constant, and the construction so changed as to increase the insulating value of the walls and roof to a theoretical case of no heat flow through them, then the ratio of the heats supplied would again be about 50 per cent, since all the heat would have to flow through the glass.

DISCUSSION

M. L. CARR (WRITTEN): This paper presents the results of the first of a series of tests made with the test houses described. A somewhat similar test was conducted in summer with cooling. Another was made with the windows of one house blanked off. The tests grew out of the tests described in a previous paper.[3]

Being a laboratory man, it is natural that I should think first of the mechanics of tests. I would call the method used an integration method. It is not a new method

[2] Radiation of Energy Through Glass, by J. L. Blackshaw and F. C. Houghten. (ASHVE TRANSACTIONS, Vol. 40, 1934, p. 93.)
[3] Heat Transfer Through Single and Double Glazing, by M. L. Carr, R. A. Miller and Leighton Orr. (ASHVE TRANSACTIONS, Vol. 44, 1938, p. 471.)

in principle, at least, and I believe has a number of things to recommend it. It also has its limitations. Considered from the standpoint of the designing engineer, it seems to me that the test results and deductions therefrom might perhaps be more convincing than laboratory results for there can be no question as to whether or not weather conditions are truly simulated. In this case, so far as over-all or end results are concerned, there is little room for questioning the validity of the ratio of amounts of heat required to maintain a given indoor temperature. One may, it is true, raise such questions as whether the test covered sufficient time, whether the season was typical, what the results of similar tests conducted in other climates would be, etc. The answers to these objections are, of course, mass data.

Probably the greatest value of such tests lies in their usefulness in checking design calculations based upon laboratory determined coefficients. For instance, the paper points out that the data seem to justify a previously expressed suspicion that the effect of wind velocity may be over-emphasized in present-day design calculations. However, it is better to over-, rather than to under-emphasize, if we must do one or the other.

As a method of determining the over-all effect of two or more constructions, for example, single and double glazing, the results obtained are very convincing, so far as those particular structures are concerned. It would, however, be an expensive way of determining the efficacy of every feature of construction design affecting heating and cooling requirements. Also, when it comes to attempting to determine the effects of wind, or of sunshine, individually upon heat loss, the method has distinct limitations. If someone can devise a way of preventing wind from blowing against one house while it is blowing against an identical test house subject to the same weather conditions, it might seem at first glance that the desired answer would be forthcoming. But, if we will pause long enough to consider, it will at once be apparent that the results will be rigorously true only for houses of the same shape and orientation and when exposed to the same ambient weather conditions other than wind. We at once begin to wonder what the difference might be if the weather were cloudy instead of bright or if it were raining or snowing. Conceivably, the effect of wind upon heat loss might be affected by such influences. An investigator is certainly privileged, if not duty-bound, to ask all such questions that his intuition or his imagination can suggest. Similar remarks apply to sunshine vs. no sunshine, though, we can compare the records of cloudy days with bright days and perhaps night records with day records. Here too, though, we at once begin to wonder as to the effect that intensity of sunshine might have.

The casual reader of the paper looking at Fig. 4, the log of the test data, probably has little conception of the amount of work involved in working up the data from the test into usable form. With the exception of the readings of the integrating watt-hour meters, all data were obtained from tapes or disk charts of recording instruments. At first thought, it may seem that this is an easy way of conducting a test and it is, so far as the recording of data is concerned. Modern recording instruments have been so perfected that, if given a little intelligent supervision, they possess many advantages over manual recording methods. They do not forget or make mistakes and they work while you sleep. But, the data they record must be condensed to be studied. This means that in a test running even for only three months, as did this one, yards of recorder tape and dozens of disk charts must be gone over and either planimetered or ordinates measured at time intervals in order to arrive at average values. Because of lag effects, very little use can be made of simultaneous, instantaneous readings.

It might seem that to determine the effect of wind for example, one need only to go over the records, select two days, one when there was considerable wind and one when it was calm. In a period, even as short as three months, probably several such

days of each kind could be found. But when one compares the total heat requirements of those days, he might not find differences of the order that other considerations would lead him to expect. If the days were bright days, then it would be natural to suspect that sunshine intensity was different. So he would then eliminate from consideration all but those days with approximately the same sunshine records. After doing this, results still might not look reasonable. By look reasonable, I mean higher heat loss on a calm day than on a windy day being compared with it. Then, in search for an explanation of this apparent discrepancy, it might be found that the days selected had been preceded by weather conditions differing greatly. We would begin to wonder if the heat capacity of the test house walls and thermal lag might not be factors. It would be possible to go on to much greater length in this vein but enough has been said to indicate that the method has its limitations as well as its good features.

It is a method that is growing in popularity, and deservedly. Also, that it is far from being perfected. Much can be done in reducing labor in working up test data, or better perhaps, in devising ways of integrating results as tests progress. I do not believe it beyond the bounds of possibility to design and build an instrument or a machine into which we could feed a temperature recorder tape and read off a dial, average hourly or daily temperatures, or, perhaps, even print them on adding machine tape. Or, perhaps recorders themselves can be devised which will record hourly or daily averages as well as instant-to-instant fluctuations.

W. C. RANDALL: With reference to Fig. 5, I would like to know the relation between mean temperature difference and the heat loss factor. In a paper by the same authors some time ago I think the conclusion was reached that the heat loss factor increased with the mean temperature difference. I would gather that Fig. 5 shows a straight line relation. I would just like to get that point cleared up because in some of our tests[4] there was no variation with mean temperature difference insofar as heat loss factor is concerned.

In connection with Fig. 7, some of our tests indicated that the effect of the velocity of the wind on heat loss was not a straight line function beyond about 8 mph. The way a lot of those points drop below the curves (beyond 8 mph—Fig. 7) might indicate that it is not a straight line ratio but drops off, and that would be very interesting if true.

So far as conclusion 3 is concerned, I think the statement which is made is correct. However, if you are interpreting the results in terms of value of double glazing versus single glazing in a home, the typical home would probably have a ratio of window area to floor area in the magnitude of 10 to 25 per cent rather than in the magnitude of 70 per cent.

Therefore, a statement, if it was to cover homes, would be a little closer to being accurate if it took into account the more general relation of window area to floor area.

W. R. RHOTON: I would like to see this kind of work carried out, particularly the possibilities of using the double-glazed glass in greenhouse work. It is one of the things that we engineers would like to know more about. In greenhouse work, as many of you know, we consider the maximum load at around 20 deg. It seems to be not any greater pull on the boiler plant when it goes to 20 below zero. It seems to hit the maximum on a 20-deg outside temperature. In order to do work along this line, it would be very helpful to the greenhouse people.

I would say that we should not carry out test houses similar to those that are

[4] Room Surface Temperature of Glass in Windows, by J. E. Emswiler and W. C. Randall. (ASHVE TRANSACTIONS, Vol. 42, 1936, p. 181.)

carried out in this experiment. I have helped do some of this work. We found out, using a ceiling type unit heater, or convected heat, which might rub the glass, increased our heat losses beyond the possibility of using that type of heat input. We found it quite necessary, where we wanted to try to raise leaf lettuce, for instance, which required air motion, to use convected heat. In that particular case we had our differential between floor and ceiling about 5 deg. In other words, we carried around 5 deg. Our maximum air temperature through the blast coil was only 60 deg. We found that was very necessary. If we used air temperature at the outlet of the unit heater around 120, our heat losses were increased enormously due to natural convected currents rubbing the glass. We had no occasion to use the double-glazed glass.

I would say there would be great possibilities to cut down the cost in greenhouse work which represents a large industry, and it would be rather interesting for the glass people to have further research along that line. I hope they keep along present lines and give us information.

It might be amusing to know that in our coefficient factors for glass at various wind velocities, we have never been able to use them in the greenhouse work. We get three or four times more heat load than is actually used in practice. We find a very interesting thing on the solar heat effect in the greenhouse. Disregarding whether the sun comes out during the daytime, take a muggy day, yet our heat loss seems to fall off on or about the noon hour, around 1 p.m. in the northern part of Ohio, so there are a lot of interesting things there that this study would be very helpful to the majority if carried out in that line.

J. D. EDWARDS: Now that people are paying more attention to problems involving solar radiation, I would like to suggest that it would be a good idea if some place in the Society's literature one could find a curve for the spectral distribution of solar energy under various typical conditions. Such a curve[8] was published recently, and on consultation with the author I found that the curve was for the distribution of energy from an arc, not the sun at all. For people who are considering problems like this, it would be very interesting to have a curve for the spectral transmission of energy through glass.

R. A. MILLER: There are available tables and charts of the spectral transmission through almost all of the various types of glass on the market. The data have not been derived from solar energy value but are based exactly on 100 per cent incident energy of a given wave-length in question.

The question of solar transmission through glass depends largely upon what we can say about a standard solar day. From all of the various data published by Dr. Abbott and others at the National Bureau of Standards, there is no such thing as a standard solar day. It seems to be impossible to arrive at any average value for the various wave length emission of solar energy. The conditions of the air on a perfectly bright day may be with such high humidities that most of the infra-red, or a considerable portion of it, is filtered out, whereas another day which appears to us exactly the same but has very low humidities, has a very high infra-red value. The CO_2 content in the air has a decided influence upon the radiant energy arriving at the earth's surface, and in attempting to do anything with at least the ordinary laboratory apparatus, an attempt to evaluate real sun values seems to be almost impossible.

We have to, if we are to proceed on a laboratory basis, take the instruments which are available and make such correction as we can to arrive at something which approaches normal sunlight. That means that the energy of the lamp has to be

[8] Figuring Solar Heat Gains of Buildings, by William Goodman. (*Heating, Piping and Air Conditioning*, October, 1938, p. 315.)

carefully measured. We have to know exactly what the filtering medium is, what the curve of energy emission from the lamp is, and the curve of energy transmission through the filter and the whole thing becomes exceedingly complicated from the point of view of deriving data of real intrinsic value to the greenhouse builders.

Furthermore, when we come to the question of radiation through glass for greenhouse purposes, it depends largely on what type of vegetation you are contemplating. If orchids are being considered, it is necessary to get a certain amount of ultra-violet with a minimum of infra-red. If there are certain other plants, you want a maximum of infra-red with a minimum of ultra-violet, and the problem again becomes rather complicated.

The effort to supply the various types of glass which one may require will certainly be made to meet the immediate demands which are presented for a given type of product. I think that can perhaps best be illustrated by the heat-absorbing glass development of the last few years which has arisen from the demand for a glass which would transmit a maximum of light and a minimum of the invisible heat rays.

LEIGHTON ORR: The most important item in Mr. Carr's discussion is the effect of the heat capacity of the walls on the test results. Our test data were recorded in 24-hour periods measured from 11:30 of one day to 11:30 of the following day. If the temperature of the walls of the houses differed from the beginning of the test period to the end, the walls would either absorb or give off a definite quantity of heat which would have a direct bearing on the measured amount of heat supplied electrically. I made an estimate, assuming a 10 F increase in outdoor temperature, from the beginning to the end of the test period and found that the amount of heat absorbed by the walls amounted to about 15 per cent of the measured heat input. It therefore follows that as long as we use fluctuating outdoor exposure conditions for test purposes, it is impossible to prevent scattering of the points as shown in the plot of Figs. 5 to 10.

In answer to Mr. Randall's question concerning the effect of mean temperature on heat transmission, the paper previously published[6] reported on laboratory tests of single and double glazing showed a decrease in the rate of heat transfer with decrease in mean temperature. However, in the results presented in this paper, the opposite condition was observed based on actual heat input as shown in the plot Fig. 6. By using as examples the two sets of data given in the accompanying table, I think we can account for there being differences in the results from the two test programs:

	Indoor Temp. deg F	Outdoor Temp. deg F	Mean Temp. deg F	Temp. Diff. deg F	Meas. Heat Input Btu/Hour/deg F Temp. Diff.	Meas. Heat Input Btu/Hour	Assumed Solar Heat Input Btu/Hour	Total Heat Input Btu/Hour	Actual Heat Transfer Btu/Hour/deg F Temp. Diff.
Case 1	70	10	40	60	130	7800	1000	8800	147
Case 2	70	50	60	20	110	2200	1000	3200	160

The indoor temperatures for both cases were 70 F and the outdoor temperatures were 10 F and 50 F. The measured heat input values of 130 and 110 were taken

[6] Loc. Cit. Note 3.

from the plot Fig. 6 as being average values for the double-glazed test house. By multiplying these heat input values by the temperature differences, the measured heat input in Btu per hour for the house were obtained. Now if we assume a solar heat input to the house of 1000 Btu per hour in each case, the total heat loss through the walls would actually be 8800 and 3200 Btu per hour. Dividing these values by their corresponding temperature differences, 60 F and 20 F we get 147 for the first case and 160 for the second case. It seems reasonable to conclude therefore, that the apparently conflicting results were due to the heat input from the sun in the outdoor exposure tests, which was not a factor in the laboratory tests.

No. 1114

HEAT TRANSFER IN STORAGE WATER HEATERS

By D. W. Nelson * (*MEMBER*) and A. A. Rosenberg,** Madison, Wis.

This paper is the result of research sponsored by the American Society of Heating and Ventilating Engineers in cooperation with the University of Wisconsin.

THE purpose of these tests was to determine capacities and heat transfer coefficients in the common type of horizontal storage water heater. This type of heater usually consists of a horizontal storage tank with an immersed bundle of copper tubes placed horizontally near the bottom. Steam is supplied to the tubes, usually through a motorized valve controlled by an immersion thermostat which is inserted in the side of the tank. In these particular tests, attention was given to the influence of water pressure on the heating capacity.

Description of Apparatus

A photograph of the apparatus used in the test work is shown in Fig. 1, and by reference to the flow diagram in Fig. 2 a comprehensive conception of the setup of the apparatus can be obtained. The storage heater used in the tests was of the horizontal cylindrical type of heavy welded construction, 30 in. in diameter and 60 in. long, having a storage capacity of 185 gal. The heating element consisted of three parallel vertical rows of *U* tubes bent to the proper radius, and rolled into a forged steel tube plate, with a center support provided for the elements. These elements consisted of 16-gage copper tubing having an outside diameter of 1¼ in.; the outside area of the tubes gave a heating surface of 10 sq ft, on the water side. For purposes of inspection and cleaning, the tank was equipped with a manhole cover at the rear and a drain at the bottom.

The heater had two 2-in. tappings: one at the bottom for the cold water inlet, and another on the top of the tank but located at the opposite end for

* Associate Professor of Mechanical Engineering, University of Wisconsin.
** Research Fellow in Mechanical Engineering 1937-38, University of Wisconsin.

Presented at the 45th Annual Meeting of the American Society of Heating and Ventilating Engineers, Pittsburgh, Pa., January, 1939.

the heated water outlet. Valves were placed on both the inlet and outlet water piping to regulate the supply of water and also its pressure in the tank. A thermometer well was located in the supply piping in order to obtain the temperature of the inlet water. Two additional thermometer wells were placed on the outlet pipe, one directly over the outlet tapping in the heater and the

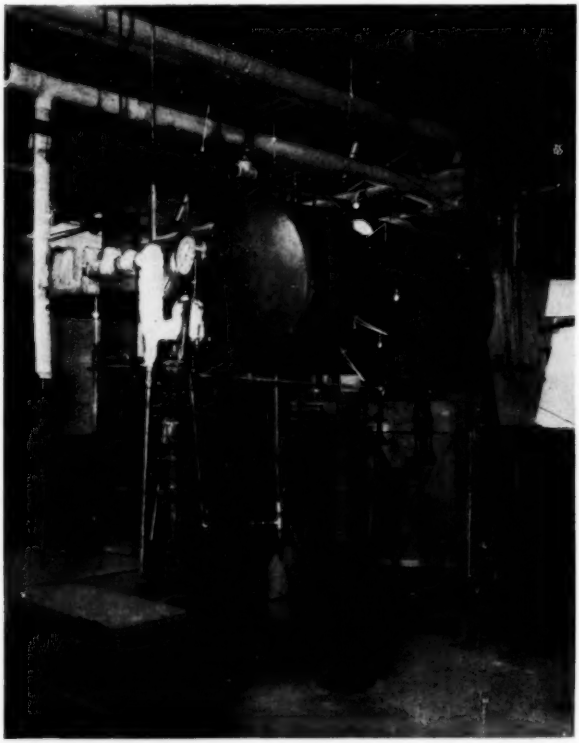

FIG. 1. PHOTOGRAPH OF TEST APPARATUS

second about a foot downstream. These served as a check on each other in reading the temperature of the outlet water.

The pressure in the tank was indicated by means of a 0 to 100 lb per square inch test gage. The water utilized for the tests was taken from the customary University of Wisconsin water supply which comes from Lake Mendota. This water is used for supplying hot water and for various needs other than for drinking, and has a hardness of about 10 grains per gallon. On one side of the tank were located six thermometer wells, as shown in Fig. 2, these serving

to indicate when the temperatures in the tank were constant during a test. A safety valve was installed in the outlet piping in order to prevent bursting of the tank, due to the expansion of water, should heat be added while the valves

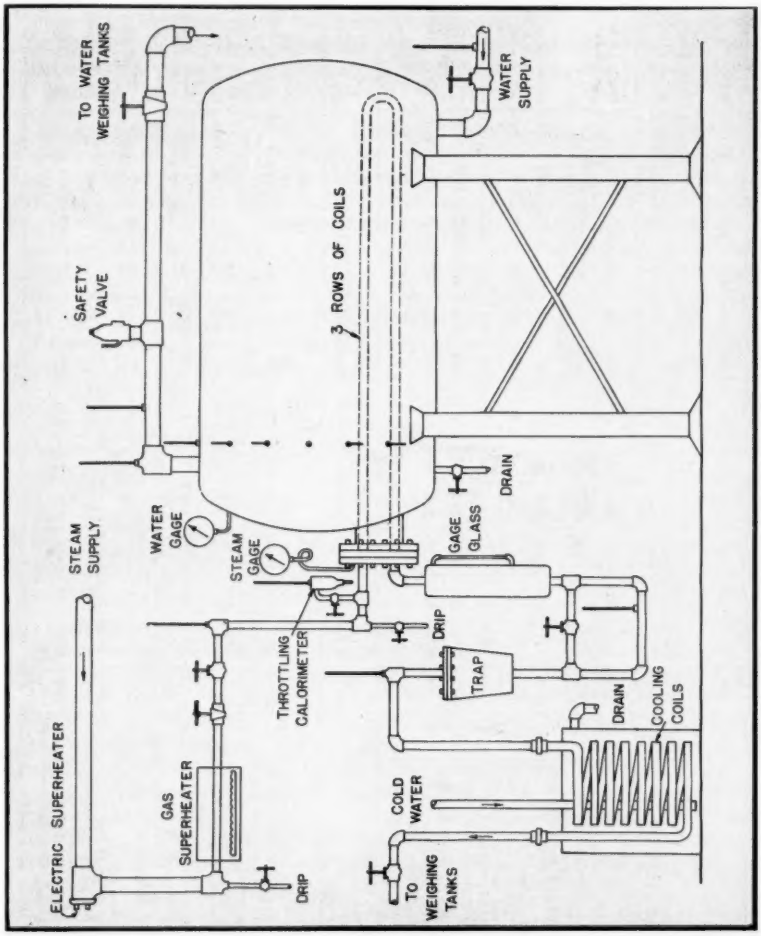

FIG. 2. DIAGRAMMATIC SKETCH OF TEST APPARATUS

were closed. The heated water was conducted through suitable piping to two large tanks of 150-gal capacity each, where the water was alternately weighed on two platform scales.

TABLE 1—ATMOSPHERIC WATER PRESSURE RUNS

	1[a]	2[a]	3[a]	4[a]	5	6[a]	7[a]	8	9[a]	10	11[a]	12[a]	13	14	15[a]	16	17	18	19[a]
1. Test Number																			
2. Steam Pressure, pounds per square inch gage	10	10	15	20	20.1	30	30	39.6	40	40	50	50	60	60	70	79.9	80	80	90
3. Quality Per Cent or Superheat deg F.	25F	36F	32F	99.8%	11F	17F	19F	2F	5F	98.0%	5F	5F	98.7%	99.0%	99.8%	99.0%	97.0%	98.8%	99.3%
4. Saturation Steam Temperature	238.5	238.5	248.5	257.8	258.1	273.4	273.5	285.7	286.2	286.3	297.3	297.2	306.9	306.9	315.7	323.6	323.5	323.6	330.9
5. Condensate Temperature, deg F.	228.2	226.4	221.7	247.0	252.8	261.1	231.2	278.2	274.9	274.8	288.9	259.3	294.8	297.5	308.1	316.4	272.1	307.5	324.4
6. Condensate Weight, pounds per hour.	578	563	746	511	719	1097	1073	963	1105	823	1474	1339	1276	1110	1777	1374	1633	1581	1959
7. Total Heat Supplied, 1000 Btu per hour	565	554	738	485	684	1041	1053	895	1031	751	1360	1275	1155	1005	1605	1220	1498	1418	1738
8. Water Temperature Outlet, deg F.	147.9	160.4	158.8	131.7	138.7	147.2	142.7	146.7	147.2	151.0	148.4	148.2	144.7	150.5	148.7	153.0	147.5	160.7	147.7
9. Water Temperature Inlet, deg F.	42.0	42.0	42.0	37.5	41.9	42.0	40.0	41.9	43.0	45.0	43.0	47.0	43.0	41.9	41.0	48.1	38.8	36.8	42.0
10. Water Heated, 100 lb per hour	53.1	46.7	64.0	51.3	71.1	99.1	102.8	85.1	100.9	69.9	129.4	125.5	113.7	90.6	149.9	117.9	138.5	114.9	164.7
11. Total Heat Absorbed, 1000 Btu per hour	562	554	747	483	688	1041	1053	890	1050	741	1363	1270	1154	984	1610	1235	1505	1420	1739
12. Radiation and Unaccounted For, 1000 Btu per hour	3	0	−9	2	−4	0	0	5	−19	10	−3	5	1	21	−5	−15	−7	−2	−1
13. Heating Capacity, gallons per hour per square foot per 100 deg rise	67.5	66.5	89.6	57.9	82.5	125.0	126.5	106.5	124.0	88.8	163.5	152.5	138.5	118.0	193.0	148.0	180.5	170.0	208.0
14. Logarithmic Mean Temperature Difference	137	128	140	169	163	175	177	187	186	184	197	195	210	206	216	219	226	219	232
15. Over-all Heat Transfer Coefficient, Btu per hour per square foot per deg F.	410	433	535	286	422	596	595	476	565	404	693	651	550	478	745	565	668	649	750
16. Water Pressure, pounds per square inch gage	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0

[a] Cleaned tubes.

TABLE 2—INTERMEDIATE WATER PRESSURE RUNS

	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
1. Test Number																		
2. Steam Pressure, pounds per square inch gage	10	20	20	20	40	40	40	40	40	40	40	60	60	80	80	80	80.1	80.3
3. Quality, Per Cent or Superheat, deg F	99.7%	3F	99.6%	99.4%	98.8%	99.2%	11F	12F	3F	10F	8F	98.5%	98.9%	98.7%	98.5%	98.6%	98.9%	99.0%
4. Saturation Steam Temperature, deg F	238.9	258.1	258.1	258.2	286.5	286.2	286.2	286.2	286.2	286.2	286.3	286.3	307.1	306.9	323.6	323.7	323.7	323.7	323.8
5. Condensate Temperature, deg F	228.1	246.5	248.4	247.9	275.9	273.5	278.9	277.9	277.7	280.5	278.7	295.0	296.9	314.5	241.0	311.5	313.0	313.5	
6. Condensate Weight, pounds per hour	311	350	348	344	454	427	1017	946	808	808	678	583	537	953	592	534	595	533	
7. Total Heat Supplied, 1000 Btu per hour	299	333	329	325	416	395	950	883	752	752	630	526	486	846	570	475	531	475	
8. Water Temperature Outlet, deg F	130.9	137.1	148.7	156.3	161.3	169.4	157.8	147.6	138.1	159.3	146.1	160.1	140.0	148.0	158.3	152.5	133.9	144.7	
9. Water Temperature Inlet, deg F	39.0	39.0	38.0	37.0	38.0	38.0	42.0	42.0	42.0	41.0	42.0	38.0	38.0	41.9	60.9	43.4	42.2	42.5	
10. Water Heated, 100 lb per hour	32.9	32.8	28.4	26.1	32.8	29.3	81.9	83.2	78.2	61.9	59.4	42.9	47.5	81.8	50.6	42.9	60.3	46.6	
11. Total Heat Absorbed, 1000 Btu per hour	302	322	314	312	405	386	946	878	751	733	620	523	485	869	494	468	553	475	
12. Radiation and Unaccounted For, 1000 Btu per hour	−3	11	15	13	11	9	4	5	1	19	10	3	1	−23	−24	7	−22	0	
13. Heating Capacity, gallons per hour per square foot per 100 deg rise	36.2	38.5	37.6	37.4	48.5	46.2	113.5	105.2	90.2	87.9	74.4	62.7	58.2	104.0	59.2	56.1	66.2	57.0	
14. Logarithmic Mean Temperature Difference	150	165	159	154	180	174	180	186	192	180	188	202	214	225	211	222	234	228	
15. Over-all Heat Transfer Coefficient, Btu per hour per square foot per deg F	202	195	198	202	225	222	526	472	391	407	330	258	227	386	234	211	236	208	
16. Water Pressure, pounds per square inch gage	25	25	25	45	25	46	5	9	12	14.3	20	25	45	20.6	55	60.3	39.6	61.1	

The steam available for the tests came through a 150-lb per square inch pressure line direct from the University Heating Station, and was brought from the underground piping tunnel first through a 2½-in. line about 90 ft long and then through a 2-in. branch about 40 ft long. The fact that this high pressure steam was generated largely from condensate from the University's heating system means that it probably contained very little air. All of the steam piping up to the inlet to the heater was covered with 2-in. pipe covering. On account of the small pipe sizes there was an appreciable drop in steam pressure, the maximum pressure available during heater tests being about 100 lb per square inch gage.

It was very desirable that the steam entering the tubes in the heater be as nearly dry as possible, or even slightly superheated. Drips were placed at two locations in the steam line before entering the heater. This was done to eliminate the possibility of any condensate being carried with the steam to the heating tubes. In order to dry and slightly superheat the steam, an ordinary gas burner was first used to heat the steam piping a short distance from the heater, as shown in Fig. 2. However, when this proved inadequate, after a number of tests, an improvised electric superheater was installed inside the 2-in. steam supply line. This consisted of nichrome wire supplied with electricity from a 220 AC source.

The flow of steam to the heater was regulated by two hand valves; the first one in the line was a gate valve, and the second was a globe valve. The latter valve served to accomplish the necessary throttling to adjust the steam to the desired pressure. This pressure was measured by means of a 0 to 200 lb per square inch steam gage; also, as a check, and to determine the superheat, if any, the steam temperature was obtained by means of a mercury-in-glass thermometer. The quality or superheat of the steam was determined just before it entered the heater. At the calorimeter, shown in Fig. 2, the temperature was read by a mercury-in-glass thermometer, and the pressure by a mercury manometer.

A receiver equipped with a gage glass was placed directly on the discharge connection from the heater. The outlet level of the condensate was then kept in sight in the gage glass during the tests, with a manually operated globe valve performing this service. In a few of the tests with higher steam pressures, it was found that the trap used was inadequate to handle the flow of condensate. Therefore, in these tests, the trap was removed and a straight section of 1-in. piping was installed; the manually operated valve alone took care of the condensate level in these instances. The condensate from the trap was cooled by passing it through a copper coil which was submerged in running cold water. This precaution was taken to prevent flashing of condensate into steam when discharged at atmospheric pressure. The condensate was alternately weighed in two 50-gal tanks placed on platform scales.

METHOD OF TESTING

The duration of a test under any one set of conditions was usually one hour. Before a test was started, the two drips in the steam line were opened, per-

mitting any condensed steam to be eliminated from the system before reaching the heating elements. After connecting the electric superheater to its source of electricity and lighting the gas burners, the steam gate valve was fully opened and the globe valve was partially opened to obtain the desired steam pressure for a particular test. The throttling valve to the calorimeter was opened to give a reading on the manometer of about 1.5 in. of mercury. With the aid of the globe valve located just before the outlet of the condensate, the level of the condensate was maintained at a predetermined point in the gage glass throughout the test.

After the heater had become warm, the water inlet valve was opened, and the supply of water was regulated to obtain the desired outlet temperature. Changing the water supply would also change the steam flow and pressure; so

TABLE 3—MAIN WATER

1. Test Number	1	2	3	4	5
2. Steam Pressure, pounds per square inch gage	10.1	20	20	20.4	40
3. Quality, Per Cent or Superheat, deg F	98.9%	99.7%	98.6%	98.7%	98.2%
4. Saturation Steam Temperature, deg F	238.8	258.1	258.4	259.0	286.3
5. Condensate Temperature, deg F	230.9	250.6	247.5	251.9	262.3
6. Condensate Weight, pounds per hour	319	317	358	385	470
7. Total Heat Supplied, 1000 Btu per hour	303	299	335	359	435
8. Water Temperature Outlet, deg F	139.8	134.4	121.7	143.1	137.6
9. Water Temperature Inlet, deg F	40.1	43.8	38.0	39.8	38.9
10. Water Heated, 100 lb per hour	29.3	32.3	39.8	31.5	39.9
11. Total Heat Absorbed, 1000 Btu per hour	292	292	333	325	394
12. Radiation and Unaccounted For, 1000 Btu per hour	11	7	2	34	41
13. Heating Capacity, gallons per hour per square foot per 100 deg rise	35.0	35.0	39.9	39.0	47.3
14. Logarithmic Mean Temperature Difference	143	165	175	162	194
15. Over-all Heat Transfer Coefficient, Btu per hour per square foot per deg F	20.4	177	190	201	203
16. Water Pressure, pounds per square inch gage	74.3	70	71.3	77.3	75.6

the two valves had to be operated simultaneously. Constancy of readings over a period of 15 min at each of the six surface tank thermometers and of the outlet water temperature served as the criterion for the attainment of equilibrium. A water outlet temperature in the range of 140 to 160 F was chosen for all the tests. The best results were obtained if conditions were fully adjusted before starting the run, for it was found that adjustments during the test resulted in fluctuations of steam and water pressures, temperatures, and weights. The two pressure gages and the four platform scales were properly calibrated and *stem corrections* were applied to the thermometer readings.

After the desired equilibrium conditions had been obtained, the run was started, and time was recorded by means of a stop watch. Two observers usually conducted the tests, one weighing the water and the other the condensate. Sometimes a third observer assisted in obtaining the various temperature and pressure readings and in regulating valves. At intervals of 10 min throughout the hour of test, the following data were recorded: temperature of water inlet, temperature of water outlet at two points, temperature of steam

at two points, temperature and pressure of the throttling calorimeter, pressures of both steam and water, temperatures of condensate before and after the trap, and the weights of both water and condensate. In some of the tests, the quantity of water was of such magnitude as to necessitate the determination of water weights every 5 min. The barometer for each run was also recorded.

Three major series of tests were made: water in the tank under atmospheric pressure (0 lb per square inch gage), water at main pressure (67 to 77 lb per square inch gage), and water at intermediate pressure (between main and atmospheric pressures). In these three series, the steam pressure was varied from 10 to 100 lb per square inch gage, as shown in the summary of results in Tables 1, 2 and 3. In Figs. 3 to 7 inclusive which show the results graphically, circles designate runs under atmospheric water pressure; squares, runs

PRESSURE RUNS

6	7	8	9	10	11	12	13	14
40	60	60	60.4	80	80	80.1	80.4	101.8
97.5%	98.7%	96.9%	98.1%	98.7%	97.3%	98.1%	98.4%	98.6%
286.1	307.1	307.1	307.2	323.5	323.7	323.7	323.7	338.8
274.0	290.9	295.3	295.0	311.5	311.3	296.9	311.5	325.8
396	541	477	580	605	604	613	622	649
360	492	424	521	540	531	552	553	570
151.2	157.2	153.7	147.3	144.5	171.2	150.7	140.9	141.1
38.0	61.3	40.0	38.9	42.9	40.0	30.9	40.1	40.9
30.4	50.3	36.6	47.9	50.9	39.3	46.1	51.9	54.4
344	482	416	519	517	515	515	523	545
16	10	8	2	23	16	37	30	25
41.3	57.8	49.9	62.3	62.0	61.8	61.8	62.8	65.2
186	195	206	209	225	212	224	230	245
185	247	202	248	230	243	230	223	222
72.5	75	66.9	71	73.8	70	71.5	75.4	74.3

under main pressure; and triangles, runs under intermediate water pressure; filled-in-circles represent those runs that were made after the tubes had been partially cleaned. This cleaning was done by removing the manhole cover, opening the drain, and then directing a stream of high pressure water on the tubes and on the inside of the tank. The substance removed from the circular bottom of the tank was a rust colored sludge about ½ in. in thickness; a thin coating of soft material was easily removed from the tube surfaces.

INTERPRETATION OF TEST RESULTS

The principal items recorded for the tests are shown in Tables 1, 2 and 3, and are shown graphically in Figs. 3 to 7 inclusive. In Fig. 3 the capacity in gallons of water heated per hour per square foot per 100 F rise is plotted against steam pressures in pounds per square inch gage. The two upper curves represent the runs made under atmospheric water pressure, whereas the lower

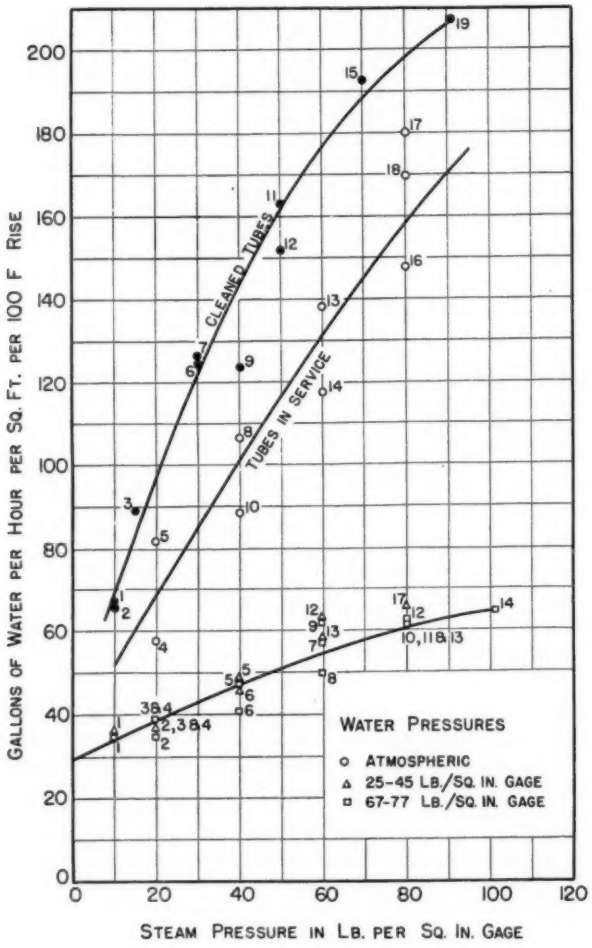

FIG. 3. CURVES SHOWING RELATIONSHIP BETWEEN WATER HEATING CAPACITY AND STEAM PRESSURE

curve is for water under higher pressures. Since the points representing intermediate water pressures of 25 to 45 lb per square inch and higher pressures of 67 to 77 lb per square inch gage fall on the same curve, it is evident that the capacity does not vary materially with the water pressure through this range.

Regardless of the amount of water pressure, as the steam pressure increases, the capacity increases; but the increase is at a much greater rate in the case

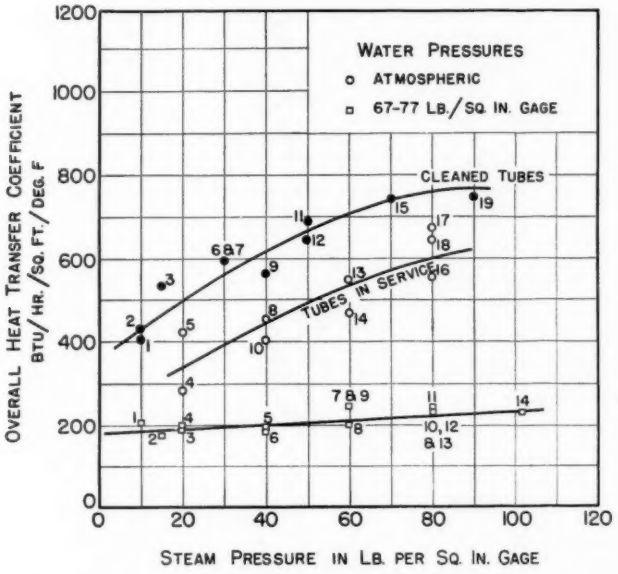

FIG. 4. OVER-ALL HEAT TRANSFER VARIATION WITH STEAM PRESSURE

of the atmospheric water pressure runs. At the higher water pressures the capacity increases from 38 to 63 gal per hour, as the steam pressure increases from 20 to 90 lb gage, or an increase in capacity of 66 per cent. Through this same range of steam pressures for the atmospheric runs, the capacity increases from 69 to 170 gal per hour, or an increase of 146 per cent. At 20 lb steam pressure the capacity at atmospheric water pressure is 1.8 times as much as at the higher water pressures; and for the 90 lb steam condition, the capacity is 2.7 times as much.

All of these results were obtained from tubes operating under a condition that might well be encountered in service. With clean tubes, it would be expected that the capacity would increase somewhat, as shown by the upper-

most curve on the sheet. After cleaning the tubes, the capacity at 20 lb steam pressure increased 40 per cent, while the capacity at 90 lb steam pressure increased only 21 per cent. Hence, it would appear that the cleaning of tubes

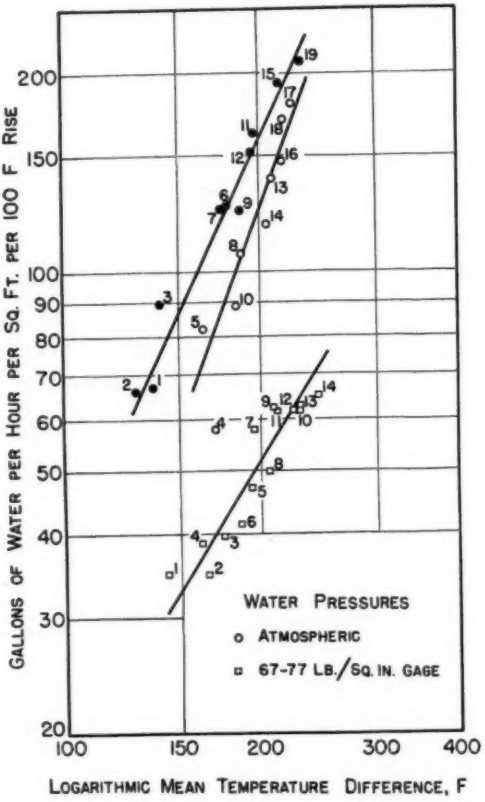

FIG. 5. VARIATION OF CAPACITY WITH LOGARITHMIC MEAN TEMPERATURE DIFFERENCE PLOTTED ON LOGARITHMIC COORDINATES

would proportionately have a greater effect on increasing the capacity at the lower than at the higher steam pressures.

The over-all heat transfer coefficient changes slightly with steam pressure, as shown in Fig. 4, when the water in the tank is maintained at the higher water pressures of 67 to 77 lb per square inch gage, with steam ranging in pressure from 20 to 90 lb the coefficient increases from 190 to 230 Btu per hour

per square foot per degree Fahrenheit, or an increase of 21 per cent. Under atmospheric water pressure, for tubes in service, this increase is from 330 to 620 Btu, or an increase of 88 per cent. At 20 lb steam pressure, the lower water pressure gives a heat transfer 1.74 times as much as for the main water

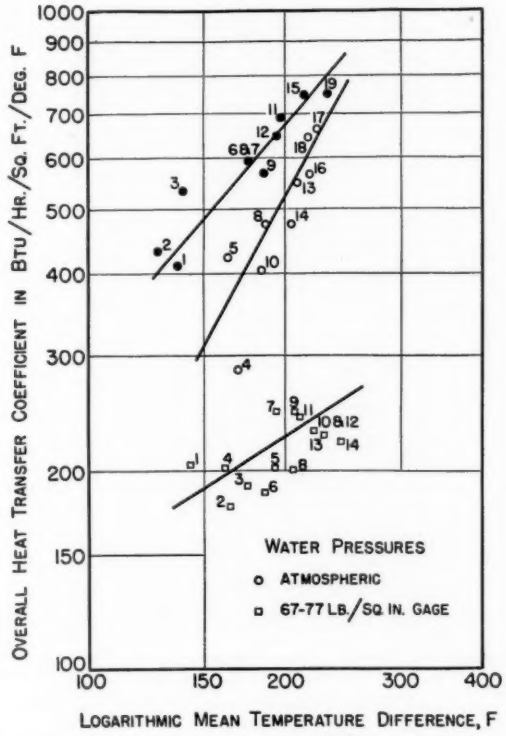

FIG. 6. CURVES SHOWING OVER-ALL HEAT TRANSFER COEFFICIENT AGAINST LOGARITHMIC MEAN TEMPERATURE DIFFERENCE

pressures; and for the higher steam pressure, the ratio is 2.7 to 1. It will be noted that these ratios for the over-all heat transfer coefficients are the same as those for capacities mentioned under discussion of Fig. 3, which naturally follows from their interdependency. For cleaned tubes and atmospheric water pressure the coefficient varies from 500 to 770 Btu per hour per square foot per degree Fahrenheit, as the steam pressure is increased from 20 to 90 lb per square inch gage.

The variation of the capacity with the logarithmic mean temperature differ-

ence under the two different water pressures is drawn to logarithmic scale in Fig. 5. This figure shows that the water pressure in addition to the mean temperature difference has a large influence in determining the capacity. Under the higher water pressures the capacity varies from 33 to 64 gal of water per hour per square foot per 100 F rise as the temperature difference increases from 150 to 230 F; whereas under atmospheric pressure the capacity increases from 60 to 178 gal through the same range. In the former instance,

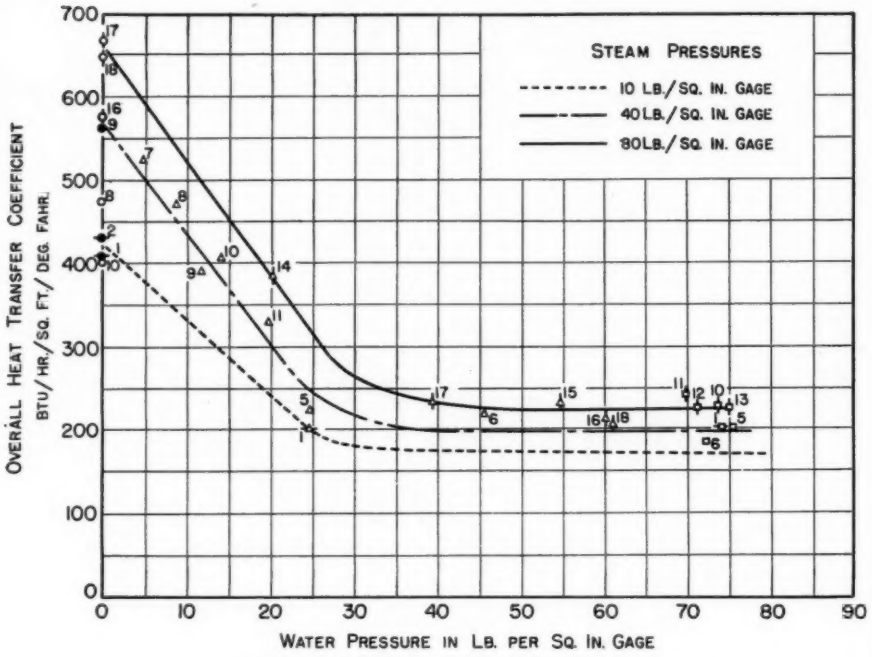

FIG. 7. GRAPHICAL RELATION BETWEEN HEAT TRANSFER COEFFICIENT AND WATER PRESSURE

the increase is about 94 per cent; in the latter, about 197 per cent. At the lower mean temperature difference the capacity is 1.8 times as great for the lower water pressure, while it is 2.8 times as great at the upper temperature difference. Cleaned tubes give a still greater capacity.

Fig. 6 is quite similar to Fig. 5 except that the over-all heat transfer coefficient is plotted against the logarithmic mean temperature difference. The large difference that results from the lower water pressure may be readily noted, for the coefficient increases at a much faster rate for the atmospheric pressure runs. With the lower water pressure the coefficient increases from about 310 to 670 Btu per hour per square foot per degree Fahrenheit through

the mean temperature difference range of 150 to 230 F, whereas this increase is only from about 190 to 240 Btu with water under main pressure.

The heat transfer coefficient is plotted against water pressure in Fig. 7. Curves are drawn for steam pressures of 10, 40 and 80 lb per square inch gage. The symbols with vertical lines through them represent the runs with 80 lb

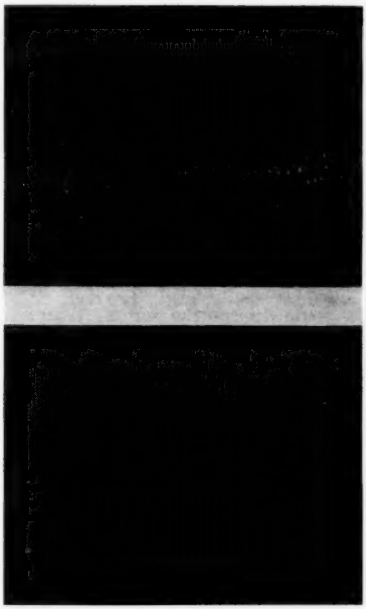

FIG. 8 (*above*). PHOTOGRAPH OF SMALL-SCALE HEAT TRANSFER APPARATUS SHOWING EFFECT OF ATMOSPHERIC WATER PRESSURE

FIG. 9 (*below*). PHOTOGRAPH OF SMALL-SCALE HEAT TRANSFER APPARATUS SHOWING EFFECT OF MAIN WATER PRESSURE

steam, while those with horizontal lines represent the lowest steam pressure. This figure indicates that for all steam pressures the coefficient and consequently the capacity are not affected as the water pressure is decreased from 75 to about 30 lb per square inch gage. However, as the water pressure decreases from 30 lb to atmospheric pressure, the coefficient increases rapidly. For the lowest of the three steam pressures, the coefficient increases 133 per cent and for the intermediate and highest steam pressures the coefficient increases 154 per cent.

The results clearly show that the capacity of a water heater of this type depends to a considerable extent upon the water pressure in the vessel, the lower pressure giving the greater capacity and heat transfer. Therefore, it was thought desirable that a qualitative explanation be found for this phenomenon; that perhaps a greater turbulence at the lower water pressure was causing the increased heat transfer. One method attempted to prove this phenomenon was to insert a light propeller in the tank to indicate the speed of circulation. The propeller was located in the tank directly over the tubes, and was connected to an indicator outside of the tank by means of a ⅛-in. diameter shaft. However, the motion in the tank was not great enough to overcome the friction of the system. Another proposal considered the insertion of a window in the side of the tank, but this was thought to be unsatisfactory for actually observing the action inside the tank near the tube surface.

A third idea, which proved satisfactory, was the duplication on a small scale of conditions inside the heater. The apparatus consisted of a ⅜-in. diameter copper tube placed inside a 1-in. diameter glass tube, the length of these tubes being about 20 in. Steam from the supply line to the heater was allowed to flow through the copper tube, which had a 0 to 50 lb pressure steam gage on the inlet end and a steam trap on the outlet. Water was conducted through suitable piping from the bottom of the heater tank to the annular space between the copper and glass tubes. By throttling with a valve on the outlet end of the apparatus, the water pressure existing in the annular space was varied. It was noticed that when the water was under atmospheric pressure, large bubbles formed on the surface of the copper tube and rapidly left the surface; whereas when the water was at a greater pressure, no bubbles were observed. Photographs taken under these two different operating water pressures are shown in Figs. 8 and 9, the former being for atmospheric water pressure and the latter for the greater pressure.

Of course it is reasonable to expect that inside the heater not all the cold water will flow directly past the tubes; probably a large part of the water is heated by its mixing with heated lower layers of water, and from steam bubbles in the circulation. This probably accounts for the fact that an over-all coefficient of heat transfer for a heater is more frequently used rather than individual coefficients. Undoubtedly the water leaving the tube surface will be very close to its boiling temperature with some tendency toward the formation of steam bubbles. It seems reasonable to suppose that the tendency of these bubbles to form and aid circulation will be influenced by the pressure of the surrounding water; the less the pressure surrounding the tubes, the greater is the tendency for the formation of bubbles and the creation of turbulence that would increase heat transfer.

CONCLUSIONS

The over-all heat transfer coefficients and consequently the capacities of storage water heaters depend to a considerable extent on the water pressure, varying inversely with the pressure.

The capacity was found to remain constant with water pressure variations in the range from 80 lb to 35 lb per square inch gage. Below a pressure of approximately 35 lb per square inch gage, the capacity increases considerably, being highest at the lowest water pressure. This difference due to water pressure is greater at high steam pressures than it is at low steam pressures.

With steam at 20 lb per square inch gage pressure the heat transfer coefficient and the capacity are about 1.8 times as great at atmospheric water pressure as at pressures above 35 lb per square inch gage. With steam at 90 lb per square inch gage the coefficient and capacity are about 2.7 times as much at the lower water pressures as at the higher water pressures.

The coefficient increases from 310 to 670 Btu per hour per square foot per degree Fahrenheit for the lower water pressure and from 190 to 240 Btu per hour per square foot per degree Fahrenheit for the higher water pressure through the range of logarithmic mean temperature differences of 150 to 230 F.

Cleaning the outer surface of the tubes increased the capacity by 40 per cent at 20 lb per square inch gage steam pressure and 21 per cent at 90 lb steam pressure, with the water under atmospheric pressure.

Visual evidence shows that the higher heat transfer at the lower water pressures is the result of the formation and movement of steam bubbles. These are formed on the tube surfaces, break away and carry heat into the main body of water above the tubes.

For tubes that had been in test service for a short time, these tests indicate a capacity of 30 gal per hour through a range of 40 to 140 F per square foot of heating surface with atmospheric steam pressure and a corresponding capacity of 51 gal at 50 lb per square inch gage. These values compare with 38.5 at atmospheric and 73 gal at 50 lb steam pressure quoted in a storage heater catalog.

ACKNOWLEDGMENT

The results in this paper were obtained in the Department of Mechanical Engineering under the general direction of Prof. G. L. Larson, chairman. Acknowledgment is due J. C. White, power plant engineer of the State of Wisconsin, for his cooperation and also to the following individuals who assisted with the tests: W. F. Gother, E. J. Rosecky, A. J. Thom, E. A. Utecht and R. A. Wilson.

DISCUSSION

PROF. G. L. TUVE (WRITTEN): There are many divisions of the field of heat transfer in which our knowledge is very meagre, and we welcome the contribution of Professor Nelson and Mr. Rosenberg, because it calls our attention to several little known facts about steam-coil water heaters.

Fig. 7 shows a rather unexpected phenomenon in the sudden break in the curves at 25 or 30 lb water pressure. Perhaps this marked difference between the high and

the low water pressures is explained by the glass tube photographs of Figs. 8 and 9, but one is led to speculate as to what would be learned by making more tests at or around the critical water pressure.

I should like to ask some specific questions:

1. Was there any opportunity for air to collect in the steam tubes?
2. Were the over-all coefficients corrected for radiation from the shell?
3. How much variation was found in the readings of the thermometers located around the circumference of the drum?
4. Was any effort made to operate the water-side at a vacuum?

In the absence of a bibliography and a correlation with other data, one might express the hope that a follow-up would be made to show whether these coefficients were typical of other design of storage water heaters.

If the steam-side surface coefficient is 2000 and the water-side coefficient for a high-pressure heater is nearer 200, are there any simple design changes which would greatly increase the over-all coefficient?

COL. W. A. DANIELSON: This is quite out of the ordinary in that nothing was provided to break up the velocity of the incoming water. That might have an important bearing on this test.

E. W. RIETZ: We are very much interested in this matter of heat transfer in storage water heaters from the temperature control angle. We have been called upon from time to time by our customers to assist them when they have not been able to get sufficient hot water or water at the temperature that they desire from their storage tank heaters.

We have been able to help them in a number of cases and we did this by changing the cold water connection to the storage tank. Instead of bringing in the cold water at the bottom, which is common practice, we changed the piping so that the cold water would enter the storage tank approximately 4 in. below the top of the tank. With this arrangement we were able to break up considerably the stratification within the tank and we eliminated the layer of very cold water at the bottom of the tank and the excessively hot water at the top, establishing a fairly uniform temperature throughout the tank, thus increasing the storage correspondingly.

We are interested to know if anybody else has tried the arrangement and what success they have had.

COLONEL DANIELSON: My understanding of breaking up that velocity is to get stratification so that there will be a body of hot water near the top quickly, and ordinarily if the water does not enter in a downward direction a plate is put over the incoming water line to break up and get the velocity stratification. I regret that these authors did not give the curve on temperature, because it has been a doubt in my mind that we have any reliable data relative to the location of the thermostatic element in the tank.

H. M. NOBIS: The arrangement of the tubes within the bundle, were they square on top of each other or at an angle, because if you lay down one tube on top of each other naturally that has a decided influence on the heat transmission, whereas if you set them diagonally you have a far freer flow of the steam bubbles which are formed, especially here, and, as the illustrations show, under low pressure water conditions. The figure does not make that statement.

PROF. D. W. NELSON: As far as we could tell by looking over the available literature, no previous tests had been published on the effect of water pressure. We

had not contemplated on specially investigating that point, but when we found the lack of data on it after happening to make some tests with atmospheric pressure in the tank and found the heat transfer so much higher than when the water was under pressure, we decided to go on with that series of tests.

One question was asked about the outlet temperature of the water in the various runs. The water temperature at the outlet of the heater was kept the same in every case.

Mr. Nobis asked a question on the tube bends. There were three tube bends arranged so that the tube went across the length of the heater and back and the three were placed in parallel. We know that the arrangement of the tubes would have a great effect on the results, and you could make a very large study just on that one point. Manufacturers of this type of heater, I think, believe that the placing of the tubes in a compact bundle will give better results than having them widely separated, because higher velocity convection currents will be set up in the body of water in the heater.

We realize that we should have had more runs near the critical pressure where the bends occurred in the curves on Fig. 7, and it is only because of lack of time and money for this project that additional points were not secured. The work continued over a period of several years, and a great many tests were run that were not shown in this report.

We made a special effort to keep air out of the steam tubes and we believe that it was not a factor in the results. An air vent was located at the inlet of the heater tube and we believe air was effectively eliminated.

The shell of the heater was not insulated and, of course, we would expect some heat loss from that, equal to, as I remember, something like 3,000 or 4,000 Btu's. Since the shell of the heater was at the same temperature at all times, we do not think it had any effect on the comparative values. The heat transfer was measured on the steam side, so it was not affected by the loss from the shell.

Colonel Danielson asked about the temperature variation in the heater. During any one run we were furnishing water to the heater at a given rate, a very steady rate, which was adjusted before the test started, and the temperature at any one point remained constant. We would find that the bottom thermometer or possibly the two bottom thermometers, which were down below the level of the tubes, would show low temperatures somewhere near to the temperature of the incoming water and above that level the temperature was quite uniformly high, so that the body of water was uniformly heated above the copper tubes.

Now, if water is just standing in there, you are in effect not having a supply and discharge line, and then those temperatures will vary considerably from top to bottom. That brings up the question about where the thermostatic element in the tank should be located. I believe that it is fairly customary to put the thermostatic element near the top of the tank, but I would feel, and some tests have shown, that it is better to put it down at the middle of the tank and perhaps somewhat lower, so that you will have a greater storage capacity of hot water at the temperature set by the thermostatic element.

No tests were made with vacuum on the water side, nor were tests made with steam under vacuum conditions.

Also, we did not try slowing down the velocity of the water as it entered the heater. The water pipe coming in was a 2-in. size, so that the velocity was comparatively low, and that water, we believe, spread over the whole length of the tank at the bottom below the copper tubes, because the temperature at all points in the tank below the tubes was substantially the same.

XUM

The rate of flow across the tubes is, of course, very low, so you cannot compare these results with results obtained in condenser tests where water moves through at very high velocity.

There is considerable scattering of the points, but I think most every one has found this scattering of points in tests involving heat transfer from vapor due to the difference between drop condensation and film condensation. Whether the steam condenses in one way or the other seems to be not very easily predictable.

The results show that scale deposit decreased the capacity of the heater. That is the effect that is generally expected. There are some tests [1] published which show that scale actually will improve the heat transfer. This is explainable by the fact that a small amount of scale increases the surface area, and at the same time gives additional points for vapor bubbles to break away from the tube surface so that an increase in heat transfer will result, as found in the quoted paper. In our tests, however, we found a decrease due to the formation of scale.

[1] Heat Transfer to Boiling Liquids, by E. T. Sauer, H. B. H. Cooper, G. A. Akin and W. H. McAdams. (*Mechanical Engineering*, September, 1938, p. 669.)

No. 1115

CONDENSATION OF MOISTURE AND ITS RELATION TO BUILDING CONSTRUCTION AND OPERATION

By F. B. Rowley,* A. B. Algren,** and C. E. Lund,*** (*MEMBERS*)
Minneapolis, Minn.

If condensation develops in a building it usually comes from the vapor which is mixed with the air on the warm side of the walls. This vapor travels from points of high vapor density to points of lower vapor density and will pass directly through many types of building construction. The rate at which it may travel is considered to be in direct proportion to the difference in vapor density between two points, and inversely proportional to the vapor resistance of the materials in the path of travel. Vapor density is usually rated in terms of vapor pressure and when the vapor pressure reaches a point which corresponds to the boiling pressure of water at a given temperature, the vapor is saturated and further moisture in the space must exist as mist or fog. As is well known, the maximum vapor density, and therefore vapor pressure, for any given space decreases as the temperature is decreased, and if the temperature of the vapor in any space is gradually decreased, ultimately a point will be reached at which the space is saturated and any further decrease in temperature will cause condensation of part of the vapor within the space. (The vapor density within a space is seldom high enough to cause trouble, unless conditions exist which make it possible for the vapor to accumulate in the form of moisture or frost within the structure.)

There are, in general, two methods by which the vapor may be accumulated either as water or frost. First, certain materials of construction are hygroscopic and will absorb a considerable quantity of moisture directly from the vapor in the air. Second, as the vapor comes in contact with surfaces which are below its dew-point temperature, condensation will take place with the accumulation of water or frost, depending upon the temperature of the surfaces. These surfaces may be the inner exposed surfaces of the walls, windows, etc., or they may be interior parts of walls. The greatest difficulties usually occur from long and severe conditions of moisture accumulation in the form of frost or ice followed by a less severe condition in which the frost is

* Director, Engrg. Experiment Station, University of Minnesota.
** Asst. Director, Engrg. Experiment Station, University of Minnesota.
*** Research Engr., Engrg. Experiment Station, University of Minnesota.

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melted at a more rapid rate than it can be vaporized and carried out of the wall. There are several causes for the condition and likewise several remedies. Usually the most severe conditions of condensation are due to a combination of causes, and it is often necessary to apply more than one specific remedy.

Briefly the causes of condensation may be summarized as:

1. High vapor density carried on the warm side of the walls.
2. Allowing this vapor to come in contact with wall surfaces or interior parts of the walls whose temperatures are below their dew-point temperatures.
3. By using exterior construction of walls which may prevent vapor from leaving the walls.
4. Exterior air temperatures which will maintain the interior parts of walls below the dew-point temperature of the vapor in contact with these parts for extended periods.

The first cause may be controlled by the conditions under which the building is operated; the second and third causes may be taken care of in the construction of the building; but the fourth cause must be anticipated from an analysis of U. S. Weather Bureau records and met by selecting the proper types of construction and reasonable operating conditions. The remedies are obviously, first, to lower the relative humidities within the building during cold periods, and second, to provide some type of construction which will prevent the moisture from travelling into the structure and which will make it possible for that moisture which does enter to be vented to the outside. Every case that presents a problem requires an analysis to determine its causes and remedies, and it is not logical to hold any one factor, such as conditions carried within the building or a certain type of building construction, as responsible for condensation until after all of the factors involved have been investigated.

An investigation covering the causes of condensation, the extent of the problem, its effect on building construction, and proper remedies to be applied has been in progress at the University of Minnesota, Engineering Experiment Station, for the past two years. A report[1] describing the test set-up and some preliminary results was given at the Annual Meeting of the Society in January 1938. Several phases of the problem have been solved, but as some of them are inter-related the problem requires additional study. This paper, therefore, must be considered as a second progress report.

During the past year the investigation has followed mainly five lines:

1. A further study of vapor barriers.
2. Ventilation of walls through the exterior surfaces.
3. The effect of vapor barriers on the drying of wet plaster.
4. The effect of attic ventilation on the accumulation of moisture and frost within the attic, and upon attic temperatures.
5. The effect of vapor pressure on the rate of vapor travel through materials.

TEST SET-UP

The complete test set-up was described in the paper previously referred to. Briefly, it consists of an insulated test room 30 ft square and 28 ft high, provided with cooling equipment by which the air temperatures can be main-

[1] Condensation Within Walls, by F. B. Rowley, A. B. Algren and C. E. Lund. (ASHVE TRANSACTIONS, Vol. 44, 1938, p. 95.)

tained continuously as low as −25 F. Small test houses are placed within this room and supplied with conditioned air, the temperature and humidity of which can be accurately controlled to match any conditions which may be met in the field. These test houses are provided with means for inspecting the interior of the walls and for measuring the effect of moisture or frost accumulation during periods of test.

The object has been to simulate as nearly as possible conditions which may be met in the field and to study the causes, effects, and corrective methods for moisture and frost conditions within structures. It has been possible to simulate the conditions insofar as the construction of buildings is concerned, and weather conditions have been duplicated with the exception of the effect of sunshine on a building. It is recognized that the periodic heat of the sun on the surface of a building may have a marked effect on the problem. In most cases, however, the effect of the sun's heat is to alleviate the difficulty, and the conditions set up in the test room are therefore more severe than those met in the field. Any remedies which will take care of test room conditions should be more than safe for equal construction and temperature conditions occurring in the field.

VAPOR BARRIERS

Two methods have been used in studying the comparative values of different types of vapor barriers. First, these barriers have been built into the wall and tested under severe temperature and humidity conditions. Aluminum panels or other vapor-proof materials have been placed on the inside surface of the sheathing to form an effective vapor stop at this point and the accumulation of frost on these vapor-stop materials has been measured and taken as the relative rate at which vapor travels through the particular vapor barrier as built into the wall. It is recognized that this method of test gives relative values only and that the vapor or frost accumulated on the aluminum or test panel is usually more than would be accumulated on the inside surface of the sheathing. It does, however, give a practical rating of the barrier and indicates definitely what may be expected of it in field application. Second, special test panels have been constructed in which the barrier under test forms the main vapor resistance between the vapor on the warm side and the vapor stop on the cold side of the barrier. The details of this test apparatus are shown in the photograph of Fig. 1 and the line drawing of Fig. 2. The individual test panels are made up of 2½ in. of insulation lined on the warm side with the barrier to be tested, and on the cold side with a light weight aluminum panel. The barrier and panel are sealed to the test frame to prevent leakage of moisture and are each spaced about ¼ in. from the insulating material. This test panel is then subjected to a definite set of conditions, such as 70 F air at 40 per cent relative humidity on the warm side and −10 F air on the cold side. Since all the vapor is supposedly stopped at the aluminum panel, the relative humidity of the cold air does not enter the problem. The vapor pressure drop may be taken as the difference between the vapor pressure of the 70 F and 40 per cent relative humidity air on the warm side and the vapor pressure corresponding to the temperature of the surface of the aluminum panel next to the insulation. This is probably not strictly equal to the vapor

pressure drop across the barrier, but it gives the same vapor pressure difference for all materials tested by this method, and if the joints are properly sealed between the frame and the panels on each side the method should give the proper relative values for the materials tested. In this test procedure the size of each test sample is 12 13/16 in. x 13¾ in., and 16 test samples are built into walls of a small test house.

Tables 1 and 2 give the relative values of different types of barriers when tested in combination with a wall, and Table 3 gives the results of certain

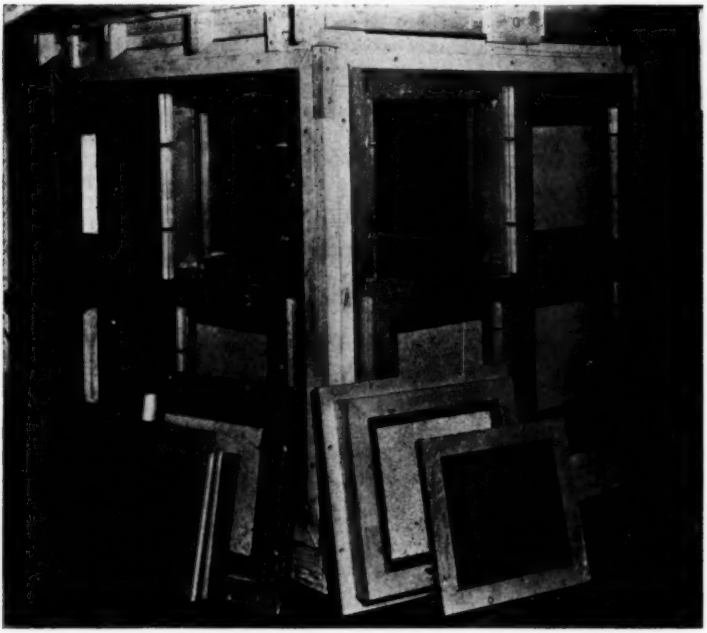

FIG. 1. VIEW OF APPARATUS FOR MAKING TESTS ON VAPOR BARRIERS

barriers when tested by the special test apparatus. The paper barriers of Table 1 were placed on the warm side of the insulation between the metal lath and studs. The paint and surface finish barriers of Table 2 were placed on the inside surface of the plaster. In all cases where test walls were used in combination with the barriers these walls were built with 2 x 4 studs spaced 16 in. on center, finished on the inside with metal lath and plaster and on the outside with 8-in. Ponderosa pine shiplap sheathing, asphalt saturated building paper, and redwood siding, with three coats of white paint. The plaster was approximately ½ in. thick consisting of a scratch coat, a brown coat, and a finish coat. The scratch and brown coats were made by mixing

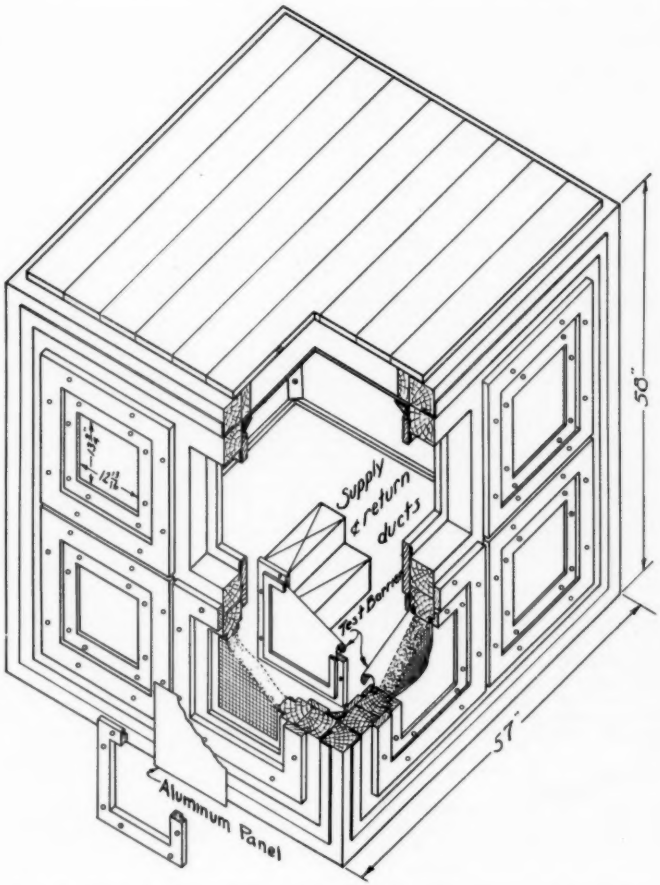

FIG. 2. LINE DRAWING SHOWING DETAILS OF CONSTRUCTION FOR SPECIAL VAPOR BARRIER TEST APPARATUS

one part of Gypsum plaster and two parts of plaster sand. The finish coat was a thin coat of lime plaster.

EFFECT OF VAPOR BARRIERS ON THE DRYING OF WET PLASTER

When an effective vapor barrier is placed underneath plaster the question may arise as to whether or not the plaster will be prevented from drying out by this barrier. If a barrier were 100 per cent effective, all of the moisture

from the plaster would necessarily have to pass out to the interior surface, whereas with no barrier it might readily pass out in both directions.

In order to answer this question a test house 4 ft wide, 8 ft long, and 8 ft high was constructed as shown in Fig. 3. The walls were built in sections

TABLE 1—CONDENSATION ON INNER SURFACE OF SHEATHING FOR DIFFERENT TYPES OF VAPOR BARRIERS PLACED BETWEEN METAL LATH AND STUDS[a]

VAPOR BARRIERS		OUTSIDE AIR TEMP. DEG F	INSIDE SURFACE TEMP. OF SHEATHING DEG F	CONDENSATION ON SHEATHING GRAMS/SQ FT/24 HOURS		
				Test No.	Test Results	Average of Tests
None	—	−19.5	− 0.3	4	2.16	—
	—	−19.5	− 1.5	4	2.26	—
	—	−19.3	1.3	8	2.02	2.15
None	—	9.9	23.0	7	1.02	—
	—	10.1	22.8	20	1.55	—
	—	9.9	22.5	7	1.65	1.41
Asphalt impregnated and surface coated glossy sheathing paper. Weight of paper tested, 51.9 lb per roll of 500 sq ft	Edges lapped not sealed	−19.5	− 4.1	4	0.09	—
		−19.5	− 2.0	8	0.05	0.07
		9.9	20.5	7	0.00	0.00
	Edges lapped and sealed	−19.5	− 3.5	4	0.02	0.02
		9.9	19.7	7	0.00	0.00
30-30-30 Duplex. Two sheets of 30 lb Kraft paper cemented together with layer of asphalt, equal in weight to 1 layer of paper. Weight of paper tested. 58 lb per roll of 500 sq ft	Edges lapped not sealed	−19.5	− 3.2	4	0.31	—
		−19.3	− 0.7	8	0.20	0.25
		9.9	20.9	7	0.00	0.00
	Edges lapped and sealed	−19.5	− 3.0	4	0.19	0.19
		9.9	20.7	7	0.00	0.00
Paper with rag felt base saturated with asphalt. Weight of paper tested. 70.8 lb per roll of 500 sq ft	Edges lapped not sealed	−19.5	− 2.2	4	0.52	0.52
		9.9	21.3	7	0.18	0.18
Duplex crepe paper. Weight of paper tested. 40.7 lb per 500 sq ft	Edges lapped and sealed	−19.5	− 1.8	4	0.09	0.09
		9.9	21.6	7	0.00	0.00

[a] Inside air conditions 70 F, 40 per cent R. H.
All walls constructed with 2 in. x 4 in. studs spaced 16 in. O. C.; metal lath and plaster on interior surface; 8 in. Ponderosa pine shiplap, building paper, and 6 in. redwood siding on outside; 3½ in. mineral wool between studs; vapor barriers as specified.

4 ft wide and 8 ft high with 2 x 4 studs spaced 16 in. on center; metal lath on the interior surface of the studs; 8-in. Ponderosa pine shiplap sheathing, building paper, and 6-in. redwood lap siding on the outside surface. The sheathing, building paper, and lap siding were built in one unit so that they could be removed as in other standard test panels. Aluminum panels were

TABLE 2—CONDENSATION ON INNER SURFACE OF SHEATHING FOR DIFFERENT TYPES OF FINISHES ON INTERIOR SURFACES OF PLASTER[a]

FINISHES ON INSIDE SURFACE OF PLASTER	NO. OF TESTS	OUTSIDE AIR TEMPERATURE DEG F	INSIDE SURFACE TEMPERATURE OF SHEATHING DEG F	CONDENSATION ON SHEATHING GRAMS/SQ FT/ 24 HOURS
Unfinished	3	−19.5	− 0.2	2.15
	3	+10.0	22.8	1.40
2 Coats Seal Coat Paint[c]	1	−19.5	0.8	0.20
	1	+10.0	23.6	0.00
2 Coats White Flat Paint[b]	1	−19.5	− 1.5	0.24
	1	+10.0	20.6	0.00
2 Coats Aluminum Paint[d]	1	−19.5	0.5	0.25
	1	+10.0	23.6	0.00
1 Coat Asphalt applied hot	1	−19.5	1.1	0.13
	1	+10.0	23.3	0.00
1 Coat Seal Coat Paint[c] 2 Coats White Flat Paint[b]	1	−19.5	1.8	0.23
	1	+10.0	22.5	0.00
1 Coat Seal Coat Paint[c] 2 Coats Aluminum Paint[d]	1	−19.5	3.9	0.16
	1	+10.0	24.3	0.00
1 Coat Glue Size[e] with Plain Wall Paper	1	−19.5	− 0.9	2.11
	1	+10.0	22.2	1.11
1 Coat Glue Size with Dull Surface Treated Canvas Wall Covering	1	−19.5	0.4	0.64
	1	+10.0	22.3	0.10
1 Coat Glue Size with Glossy Surface Treated Canvas Wall Covering	1	−19.5	0.7	0.47
	1	+10.0	21.7	0.01
1 Coat Glue Size with Duplex Crepe Paper[e]	1	−19.5	− 0.2	0.32
	1	+10.0	21.8	0.0

[a] Inside air conditions 70 F and 40 per cent R. H. All walls constructed with 2 in. x 4 in. studs spaced 16 in. on centers; metal lath and plaster on interior surface; 8 in. Ponderosa pine shiplap, building paper and 6 in. redwood siding on outside; 3⅝ in. mineral wool between studs; surface finish as specified.

[b] *White Flat Interior Paint Specifications:*
Pigment....62.3 per cent
Vehicle....37.7 per cent
Pigment Composition:
Titanium calcium pigment....78.0 per cent
Calcium carbonate....22.0 per cent
Composition of titanium calcium pigment:
Titanium oxide....30.0 per cent
Calcium sulphate....70.0 per cent
Vehicle Composition:
Vegetable oil and resin....34.3 per cent
Volatile thinner and drier....65.7 per cent

[c] *Seal Coat Specifications for Priming Coat on Plaster Surfaces:*
Pigment....40 per cent
Vehicle....60 per cent
Pigment Composition:
Titanium calcium pigment....100.0 per cent
Composition of titanium calcium pigment:
Titanium oxide....30.0 per cent
Calcium sulphate....70.0 per cent
Vehicle Composition:
Resins....16.7 per cent
Vegetable oil....28.8 per cent
Drier and thinner....54.5 per cent

[d] *Aluminum Paint Specifications:*
Aluminum paste....1.75 lb
Vehicle....7.40 lb or 1 gal
Composition of paste:
Pure aluminum powder....65.0 per cent
Volatile liquid....35.0 per cent
Vehicle:
Water resisting spar varnish.

[e] *Glue Size Specifications:*
A cold water size containing animal glue.
Mixture: 11 pints of water to 1 lb of size.
Note: *Outside White Lead Paint Specifications:*
Pigment by weight....70.0 per cent
Vehicle by weight....30.0 per cent
Composition of pigment:
A pure white lead paste.
Composition of vehicle:
Linseed oil....90.0 per cent
Drier....10.0 per cent

placed between each stud section next to the sheathing for the purpose of collecting and weighing frost which might accumulate due to the passage of moisture through the wall. An air duct was provided such that cold outside air could be circulated to the interior of the test house at a rate corresponding to two changes per hour for the average house. The walls were constructed with various types of vapor barriers placed under the metal lath, and

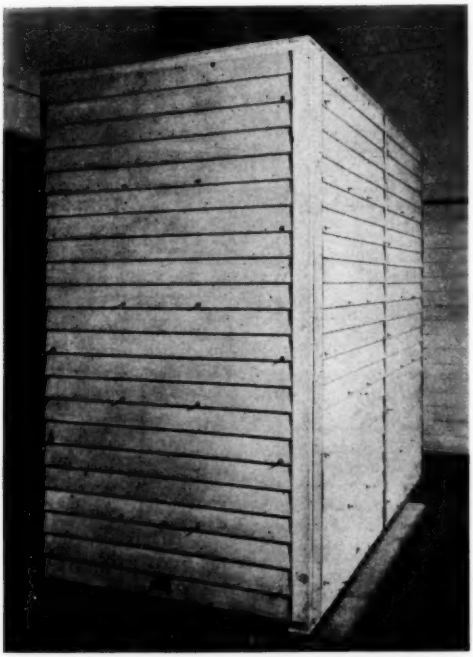

FIG. 3. TEST HOUSE USED FOR WET PLASTER TEST AND ALSO SUBSEQUENT TEST ON WALL VENTILATION

with 3⅝ in. of mineral wool applied between the studs. After the test house was constructed and before any plaster was applied, the inside and outside air temperatures were brought to 70 F and −15 F respectively, and maintained for a sufficient length of time to establish equilibrium through the construction. The plaster was then applied in three coats consisting of a scratch coat, brown coat and finish coat, each allowed to dry for the time indicated. The composition of the plaster was as given in Table 4.

The important conclusions which can be drawn from the results of the wet plaster test shown in Table 4 are:

1. A good vapor barrier applied underneath the plaster is effective in stopping the moisture travel from the wet plaster to the interior of the wall, but the relative humidity within the structure may run very high during the drying out period.

TABLE 3—AVERAGE RESULTS FOR VAPOR BARRIER TESTED BY SPECIAL TEST METHOD[a]

DESCRIPTION OF BARRIER	NO. OF TESTS	CONDENSATION GRAMS/SQ FT/24 HOURS		
		MAXIMUM	MINIMUM	AVERAGE
No barrier	1	—	—	13.40
Asphalt impregnated and surface coated glossy sheathing paper. Weight of paper tested, 51.9 lb per roll of 500 sq ft	5	0.22	0.09	0.16
30-30-30 Duplex Two sheets of 30 lb Kraft paper cemented together with a layer of asphalt, equal in weight to 1 layer of paper. Weight of paper tested, 58 lb per roll of 500 sq ft	3 2 2	0.46 0.40 0.44	0.20 0.35 0.43	0.37 0.38 0.43
Paper with rag felt base saturated with asphalt. Weight of paper tested, 70.8 lb per roll of 500 sq ft	3	2.12	1.04	1.46
Single sheet 50 lb Kraft paper, saturated with asphalt, finished weight 68 lb. Weight of paper tested, 12 lb per 500 sq ft	1	—	—	1.75
Single sheet 70 lb Kraft paper, saturated with asphalt, finished weight 82.5 lb. Weight of paper tested, 14.8 lb per 500 sq ft	1	—	—	3.00
Duplex crepe paper, weight of paper tested, 40.7 lb per 500 sq ft	3	0.24	0.11	0.16

[a] Inside air conditions 70 F, 40 per cent R. H. Outside air conditions—10 F.

TABLE 4—AIR TEMPERATURES AND HUMIDITY CONDITIONS FOR WET PLASTER TESTS

PLASTER	NO. OF HOURS	COLD ROOM DEG F	TEST HOUSE			
			D. B., Deg F	PER CENT R. H.		
				Max.	Min.	Avg.
Scratch Coat[a]	66	−15.2	69.9	73.0	57.0	63.6
Brown Coat[b]	96	−15.4	70.1	78.0	59.0	62.7
Finish Coat[c]	171	−14.3	70.0	75.0	34.0	58.0

[a] Scratch Coat—sand passing No. 10 sieve, 500 lb; fiber plaster, 204 lb; and water, 105 lb.
[b] Brown Coat—sand passing No. 10 sieve, 880 lb; fiber plaster, 353 lb; and water, 185 lb.
[c] Finish Coat—hydrated lime and unfibered plaster to ratio of approximately 3 parts of lime to 1 part of plaster mixed with water to give proper workability.
Note: The mixtures for the scratch and brown coats consisted of 2½ parts of sand by weight, 1 part of plaster, and 6 gal of water per sack of plaster.

2. A barrier should be used which will not be affected by wet plaster.

3. The drying out of the plaster does not appear to be affected by the presence of the barrier and no difference can be detected between the plaster dried with the barrier and that dried without the barrier.

4. During the drying period for plaster over an efficient barrier some moisture may accumulate on the warm surface of the barrier, but this will re-evaporate and vent out through the dried plaster without causing any damage to the structure.

VENTILATION OF WALLS TO EXTERIOR AIR

Vapor passes through all parts of a wall from points of high vapor pressure to points of low vapor pressure. It appears to travel in direct proportion to the differences in vapor pressure, and in inverse proportion to the vapor resisting property of that part of the wall through which it passes. If the conditions of vapor pressure and vapor resistance throughout a wall are such that there is a uniform and equal flow of vapor through each section of the wall, a normal moisture gradient will be established throughout the wall and there will be no point within the wall at which moisture or frost will accumulate. If, on the other hand, conditions permit the vapor to travel at a higher rate through the first section than through the other sections of the wall, there will necessarily be an accumulation of moisture or frost at the line where this rate of travel changes. In order to avoid condensation difficulties it is desirable to construct a wall, which under the conditions of use will resist the entrance of vapor through the warm interior surface, and as a factor of safety permit the free passage through the exterior part of the wall of any vapor which may have entered the interior section.

The passage of vapor from the interior part of the wall to the exterior cold air is often spoken of as ventilation of the wall, and is connected with the movement of air through the exterior surface of the wall. As a matter of fact the venting of vapor may take place without any air movement at all, although it is usually accompanied by some exchange of air. There are cases in practical construction in which it appears desirable to accelerate the normal passage of vapor through the exterior of the wall by means of artificial ventilation. Any exchange of air through the exterior surface of a wall will increase the heat loss through the wall, but under some conditions it is possible to provide enough artificial ventilation to relieve any moisture accumulation within the wall and yet not materially increase the heat loss through the wall. It is this type of ventilation that is now under investigation. Different methods of ventilating through the exterior surfaces of walls have been applied to both 4-ft and 8-ft walls with different kinds of construction. These tests are still in progress but from the work thus far done it may be said that ventilation of walls to the exterior cold air will increase the rate of vapor travel through the exterior surface and give partial or complete relief from any condensation difficulties depending upon the severity of conditions. For severe conditions it usually will be necessary to consider additional corrective measures such as reducing the inside relative humidity or using some vapor resistant material on the warm side of the wall.

VENTILATION OF ATTIC SPACE

When insulation is applied to the ceiling of the upper floor and the attic space above is unheated, the under side of the roof and attic walls may become very cold and provide a surface for the condensation of moisture and accumulation of frost, if vapor is allowed to pass through the upper floor ceiling into the attic space. The most effective way of preventing this condensation is to place a vapor barrier on the warm side of the insulation between the attic space and the rooms below. In case it is not practical to use this type of a barrier, or as a precaution against any moisture which may

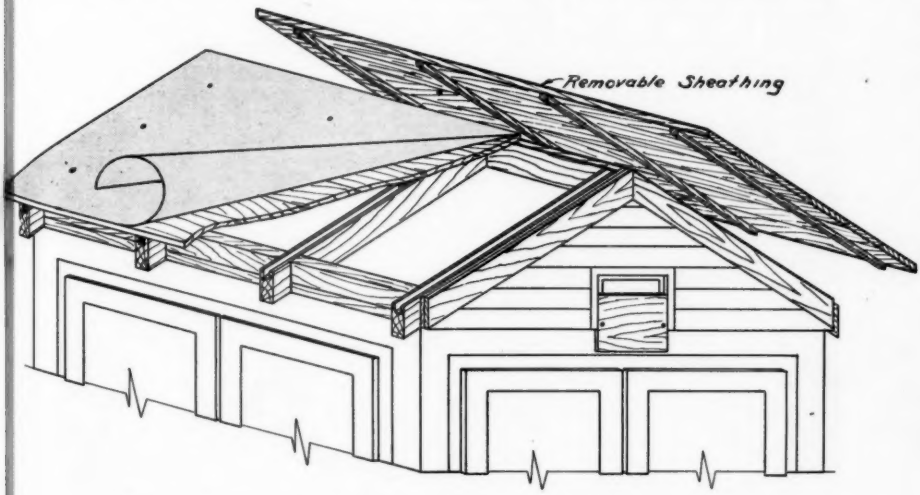

FIG. 4. CONSTRUCTION DETAILS OF ATTIC FOR ATTIC VENTILATION TESTS

pass the barrier, it is usually possible and practical to ventilate the attic space to the exterior of the building. The question then arises as to the type of ventilation system to use and the minimum amount of air necessary for effective ventilation.

In the preliminary investigation of the possibilities of attic ventilation three of the small size test houses have been equipped with attics. The ceilings below were constructed with 2 × 4 joists, metal lath and plaster, with 3½-in. mineral wool between the joists. No vapor barriers were used. The roofs were constructed with 8-in. pine shiplap sheathing covered with roofing paper. The attic spaces were typical for cold attics with cold surfaces underneath the roofs for the condensation of moisture. The first attic was completely enclosed without ventilation; the second was provided with adjustable vent openings in each gable; and the third was provided with mechanical ventilation by which the amount of air supplied to the attic could be accurately measured. In each of the attics the roof boards with attached roofing material were built as a unit so that the entire roof section could be removed during test for

inspection of interior conditions if desired. Small condensation panels were also placed on the under side of the roof sections to provide means of collecting and weighing the frost accumulated during a test period. Observation openings were provided in the gables for inspection of the interior to determine whether or not condensation was taking place during a test period without disassembling the walls. Fig. 4 shows a line drawing of the attic in which openings were provided for natural ventilation through the gables. Fig. 5

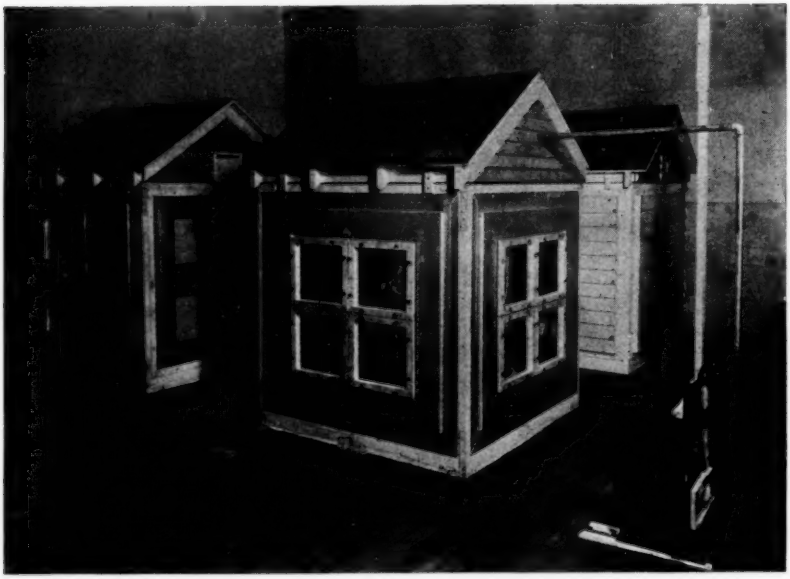

FIG. 5. VIEW OF SET-UP FOR VENTILATED AND UNVENTILATED ATTICS. APPARATUS IN FOREGROUND USED FOR MEASURING SUPPLY OF AIR TO MECHANICALLY VENTILATED ATTICS

shows the three test attics assembled in the cold room together with the pump and meter used to furnish air to the mechanically ventilated attic.

The test data thus far taken on the three attics are shown in Table 5. Tests have been run with outside temperatures ranging from −10 F to +25 F and with air conditions in the test houses ranging from 70 F and 40 per cent relative humidity to 70 F and 20 per cent relative humidity.

From the results in Table 5, test 19, period 1, which was run at a rather severe condition of 70.3 F, 40.5 per cent relative humidity for inside air, and −10.4 F for outside air, a rate of condensation on the underside of the roof for the unventilated attic of 3.01 g (grams) is shown. Under the same conditions the attic with natural ventilation having a ¼ sq in. opening on each gable per square foot of ceiling area, and the attic with mechanical ventila-

TABLE 5—ATTIC VENTILATION

TEST No.	PERIOD No.	DURATION OF TEST	OUTSIDE AIR TEMP. DEG F	INSIDE AIR CONDITIONS		NO VENTILATION		NATURAL VENTILATION			MECHANICAL VENTILATION		
				Deg F	R. H. %	Attic Air, Temp.	Condensation[a]	Opening, sq in[b]	Attic Air Temp.	Condensation	Air Supplied, cu ft[c]	Attic Air Temp.	Condensation
19	1	54	−10.4	70.3	40.5	4.6	3.01	0.25	−0.4	0.0	3.0	1.8	0.00
19	2–3	68	−10.3	70.0	40.9	4.6	3.33	0.125	1.2	0.16	1.5	2.0	0.03
20	1	66	+15.0	70.1	40.6	26.9	0.0	0.125	24.2	0.0	1.5	24.3	0.00
20	2	99	+10.0	69.9	40.3	22.6	0.0	0.125	19.9	0.0	1.5	19.7	0.00
20	3	117	+5.0	70.1	40.6	17.3	1.16	0.125	13.9	0.0	1.5	14.4	0.00
20	4	72	+5.0	70.0	31.3	17.6	0.78	0.125	14.6	0.0	1.5	15.0	0.00
20	5	72	+5.0	70.1	20.7	16.9	−1.02	0.125	14.1	0.0	1.5	14.4	0.00
20	6	98	+10.0	70.1	20.6	20.5	−1.22	0.125	18.3	0.0	1.5	18.4	0.00
20	7	66	+15.0	70.1	21.0	23.4	0.0	0.125	21.2	0.0	1.5	21.2	0.00
20	12	141	0.0	70.0	21.3	11.8	1.76	0.125	7.8	0.0	1.5	9.6	0.00
20	13	72	−11.4	70.1	19.9	1.43	2.52	0.125	−2.5	0.0	1.5	−0.5	0.00

[a] Condensation in grams per square foot ceiling area per 24 hours.
[b] Opening in square inches per square foot ceiling area in each gable.
[c] Air supplied in cubic feet per hour per square foot ceiling area.

tion having 3 cu ft of air per hour per square foot of ceiling area, showed no condensation. Test 19, period 2-3 run under the same temperature and humidity conditions gave 3.33 g condensation for the unventilated attic and 0.16 for the attic with natural ventilation with opening at each end reduced to ⅛ sq in. per square foot of ceiling area, and 0.03 g for the mechanically ventilated attic supplied with 1½ cu ft of air per square foot of ceiling area per hour. Evidently test period number 2-3 represents the minimum of ventilation for either natural or mechanical ventilated attic.

The loss of heat due to attic ventilation may be calculated for both the natural and mechanical ventilation by assuming that the loss of heat through each ceiling is directly proportional to the drop in temperature from inside air to attic air, and that all ceilings have the same conductance values. On this basis the increase in heat loss through the ceilings during period number 2-3 for the natural ventilated attic as compared with that of ventilation is 5 per cent, and for the mechanically ventilated attic it is 4 per cent. An average of all the tests thus far run seems to indicate a heat loss due to ventilation of approximately 5 per cent for the conditions given in test period number 2-3.

The second approximate method of figuring the heat loss due to mechanical ventilation would be to assume that the loss is equal to the amount of heat required to heat the ventilation air from outside temperature to attic temperature. On this basis the heat loss caused by mechanical ventilation for period number 2-3 would be the amount required to heat 1.5 cu ft per hour through a temperature range from −10.3 F to +2 F, or 0.39 Btu per square foot of ceiling area per hour. If a coefficient of heat transmission for the complete ceiling of 0.08 and a temperature drop of 65.4 are assumed as in the unventilated attic the heat loss per square foot of ceiling area for the unventilated attic is 5.23. Thus the calculated loss would be 7.4 per cent. This method would give an excessive loss charged against ventilation since it assumes that there is no loss for the unventilated attic. It seems safe to assume that the loss due to adequate ventilation would not be more than 5 per cent when based on the conductivity through the ceiling, and excepting for the most severe conditions it should be less than this amount. The data thus far obtained have been on small sized attics and of a single type of roof. The application to larger sized attics will be investigated later.

RELATION OF VAPOR CONDUCTANCE THROUGH A MATERIAL TO THE DROP IN VAPOR PRESSURE

There has been much speculation as to whether the rate of vapor travel through a material or combination of materials as in a wall is in direct proportion to the drop in vapor pressure along the path of vapor flow, or whether the rate follows some other law. It is probably true that the manner in which the vapor is conducted through various parts of the path for various combinations of material is not the same for all conditions of vapor flow. For certain cases such as impervious hygroscopic materials the vapor may be hygroscopically absorbed into one surface of the material and transmitted through to the opposite side, as a liquid saturating the material, from where it will be

re-evaporated. In other cases it may be transmitted directly through the material as a vapor without any change in state. The laws governing the rate of transfer may be affected by temperature range, method of transfer, or by some property of the material. If, however, the rate of transfer is substantially in direct proportion to the vapor pressure drop along the path of transfer, then the problem of vapor flow may be considered similar to that of heat flow through materials, and vapor conductance coefficients may be developed for individual materials and combinations of materials similar to the heat conductance coefficients now in use.

Sufficient test data have not yet been taken to prove whether or not the rate of vapor flow through the frame walls now being investigated may be

TABLE 6—TEST RESULTS INDICATING THE RATE OF VAPOR TRAVEL THROUGH MATERIAL IS PROPORTIONAL TO THE VAPOR PRESSURE DIFFERENCE

TEST AND PERIOD NO.	OUTSIDE AIR TEMP. DEG F	INSIDE AIR CONDITIONS		VAPOR PRESSURE IN. HG			CONDENSATION ON SHEATHING, GRAMS/SQ FT/ 24 HOURS	V_1[a]
		Deg F	R. H. %	Inside Air	Inside Surface of Sheathing	Outside Air		
20–1	15.0	70.2	40.7	0.3003	0.1392	0.0565	1.53	....
20–7	15.1	70.1	20.8	0.1543	0.1392	0.0568	−0.29	12.46
20–2	10.1	69.9	40.6	0.2992	0.1175	0.0442	1.55	....
20–6	10.0	70.1	20.6	0.1521	0.1175	0.0440	−0.44	13.50
20–3	5.0	70.3	40.7	0.3037	0.0980	0.0342	1.93	....
20–5	5.0	70.1	20.6	0.1521	0.0980	0.0342	0.40	10.10
20–2	−11.4	70.4	40.0	0.3016	0.0566	0.0144	2.66	....
20–13	−11.4	70.0	20.0	0.1480	0.0561	0.0144	0.97	11.05
7–4	9.9	70.0	41.0	0.3029	0.1159	0.0438	1.65	....
20–6	10.0	70.1	20.6	0.1521	0.1157	0.0440	−0.44	13.82

[a] V_1 = Calculated vapor transmittance coefficient from inside air to inside surface of sheathing in grams per square foot per 24 hours per inch of Hg pressure difference.

considered substantially proportional to the vapor pressure drop. In order to get some indication in regard to this law, test values for several walls were selected and compiled in Table 6. These test values are either from identical walls or from walls constructed from the same specifications. They were all frame walls with metal lath and ½-in. gypsum plaster on the inside surface, and pine sheathing, building paper, and lap siding on the outside surface with 3⅝ in. mineral wool between the studs. They were all tested at substantially 70 F inside air temperature with humidities of approximately 20 per cent and 40 per cent for outside temperatures of 5 F, 10 F, 15 F, and −11.4 F. The test results for these walls are arranged in the table to show the comparative values for humidities of 20 and 40 per cent at the different cold room temperatures.

Since in each pair of tests all test conditions were the same with the exception of the vapor pressure on the warm side of the wall, the rate of vapor flow from the sheathing surface to the outside air should be the same for both tests of a pair, and the difference in frost accumulation on the sheathing should be caused entirely by the difference in rate of vapor flow

from warm air to the sheathing. The only change in conditions which could cause a change in rate of flow is in the vapor pressure on the warm side, and it is reasonable to assume that this difference in vapor pressure drop between the two tests of a pair was entirely responsible for the difference in frost accumulation on the sheathing for the two tests.

Consider the first two lines of test results in the table. In this case the same wall was tested at a cold room temperature of 15 F and conditioned air on the inside of the test house of 70 F with 40.7 per cent relative humidity for the first test and 70 F and 20.8 per cent relative humidity for the second test. In this pair of tests the conditions were all substantially the same with the exception of the vapor pressures on the warm side of the test wall. These were 0.3003 in. of mercury for the first test and 0.1543 in. of mercury for the second test, giving a difference of 0.146 in. of mercury. The rate of condensation on the aluminum panel lining the inside of the sheathing was 1.53 g per square foot per 24 hours for the first test and −0.29 g for the second test, giving a total difference of 1.82 g. The rate of condensation per inch of pressure difference is therefore $\frac{1.82}{0.146}$ or 12.46. This value may be considered as the vapor conductance value in grams of moisture per square foot of area per 24 hours per inch of mercury pressure drop for the combination of plaster, metal lath and insulation up to the sheathing, and for convenience may be designated as V_1. A similar calculation for the other pairs of test values shown in the table gives the individual values as listed with an average of 12.19 for the five cases. The variation in these values is not great considering the test method used, and the results indicate that the flow of vapor is in direct proportion to the drop in vapor pressure along the path of flow.

A significant point to be noted from Table 6 is the marked effect of relative humidity of the inside air on the rate of condensation within the wall. For outside temperatures of 10 or more degrees above zero a drop in relative humidity of the inside air from 40 to 20 per cent entirely cleared up any condensation difficulties and reduced the amount of moisture previously accumulated in the wall. At 5 F outside air temperature the rate of condensation was reduced by about 80 per cent, and at −11.4 F it was reduced 71 per cent by reducing the relative humidity of the inside air from 40 to 20 per cent. From these results it is evident that high relative humidity is one of the major factors in condensation problems and one of the first points which should be investigated when condensation occurs. Relative humidities of inside air should be controlled with reference to outside weather conditions.

EFFECT OF VARIABLE OUTSIDE TEMPERATURE ON THE RATE OF MOISTURE CONDENSATION IN A WALL

In most of the tests thus far made the cold room temperature has been maintained constant throughout an extended test period and the frost accumulation in the different parts of the walls has been due to continuous uniform temperature difference between the inside and outside air. In practice the outside air temperature does not remain constant throughout long periods, and even for rather extended cold periods there is a variation in outside tempera-

ture over the 24-hour period. The daily variation of temperature may be taken as approximately 15 F, but the variation through any ordinary low temperature period will be much greater than this. The question naturally arises as to whether the condensation conditions within a wall are the same when the wall is subjected to a variable outside temperature condition as they are for the same wall subjected to a constant outside temperature condition which would be equal to the average of the variable temperatures, all other conditions remaining the same.

In order to compare the results which might be expected for a given wall operating under uniform outside temperature conditions to those for the same wall operating under variable outside temperatures having the same average as the uniform temperatures, consider a frame wall with 3⅝ in. of insulation

TABLE 7—CALCULATED TEMPERATURES AND VAPOR PRESSURES THROUGH INSULATED WALL FOR DIFFERENT OUTSIDE AIR TEMPERATURES—OTHER CONDITIONS REMAINING CONSTANT

INSIDE AIR		OUTSIDE AIR		INSIDE SURFACE TEMP. OF SHEATHING	VAPOR PRESSURE		
Temp. Deg F	R. H.%	Temp. Deg F	R. H.%		Inside Air	Sheathing	Outside Air
70	40	0	50	11.9	0.2955	0.0693	0.0189
70	40	−20	50	−4.75	0.2955	0.0321	0.0063
70	40	20	50	28.5	0.2955	0.1535	0.0514

(1) $(0.2955 - 0.0693)\ V_1 - (0.0693 - 0.0189)\ V_2$ = rate of condensation for zero degrees $0.2262\ V_1 - 0.0504\ V_2 = R_1$.
(2) $(0.2955 - 0.0321)\ V_1 - (0.0321 - 0.0063)\ V_2$ = rate of condensation for −20 F $0.2634\ V_1 - 0.0258\ V_2 = R_2$.
(3) $(0.2955 - 0.1535)\ V_1 - (0.1535 - 0.0514)\ V_2$ = rate of condensation for 20 F $0.1420\ V_1 - 0.1021\ V_2 = R_3$.
(4) Average of (2) and (3) $0.2027\ V_1 - 0.0639\ V_2$ = rate of condensation for average of two conditions +20 to −20 F.

between the studs. Consider this wall to be subjected to a test condition consisting of warm air on the inside of the wall of 70 F and 40 per cent relative humidity, and cold air on the outside of 0 F and 50 per cent relative humidity, and compare the probable result for the same wall subjected to the same inside air conditions but with an outside temperature ranging from −20 F and 50 per cent relative humidity to +20 F and 50 per cent relative humidity, thus having an average outside air temperature of zero.

Table 7 shows the temperatures and vapor pressures for the air on the warm side, the inner surface of sheathing, and the air on the cold side of the wall for the three test conditions. If it is assumed that the rate of vapor travel between two parts of the wall is equal to a constant multiplied by the drop in vapor pressure between these parts, then equations numbers (1), (2) and (3) show the rates of frost accumulation on the sheathing for the three test conditions, and equation number (4) shows the average for conditions (2) and (3) which is to be compared with test condition (1). An inspection of equations numbers (1) and (4) shows that the rate of vapor travel from the inside air to the sheathing is greater for the constant air temperature condition than it is for the variable air conditions. At the same time the rate of vapor travel from the surface of the sheathing to the outside air will be

greater for the condition of variable air temperature than for the condition of constant air temperature.

In order to check the correctness of these calculated results test data have been compiled in Table 8 for three tests, the first with a cold room temperature of +10 F, the second 0 F, and the third −10 F. All other conditions were maintained substantially the same throughout the three tests. In the last column of the table the rates of condensation are shown, first on the surface of sheathing of an insulated wall without vapor barrier, and second on the underside of the roof over an unventilated attic with insulation in ceiling below. The average rate of condensation on the sheathing of the wall for the two tests of +10 F and −10 F is 0.265 g, whereas for the test at zero degrees it is

TABLE 8—RELATION BETWEEN CONDENSATION RATE FOR CONSTANT OUTSIDE TEMPERATURES AND FOR VARIABLE OUTSIDE AIR TEMPERATURES FOR WALLS AND ATTICS

Test and Period No.	Outside Air Temp. Deg F	Inside Air Conditions		Vapor Pressure In. Hg			Condensation Rate on Sheathing Grams/sq ft/24 Hours
		Deg F	R. H.%	Inside Air	Outside Air	Inside Surface of Sheathing	
Walls							
20–6	+10.0	70.1	20.6	0.1521	0.0440	0.1175	−0.44
20–12	0.0	70.1	20.5	0.1514	0.0266	0.0815	0.64
20–13	−10.0	70.0	20.0	0.1480	0.0144	0.0561	0.97
Attics							
20–6	+10.0	70.1	20.6	0.1521	0.0440	–	−1.22
20–12	0.0	70.1	20.5	0.1514	0.0266	–	1.76
20–13	−10.0	70.0	20.0	0.1480	0.0144	–	2.52

0.64 g per square foot of sheathing surface per 24 hours. For the underside of the roof over the attics the average rate for the two tests of +10 F and −10 F is 0.65 g as compared with 1.76 g per square foot of ceiling area at the zero degrees temperature. These test results bear out the calculated results as taken from figures of Table 7. It should be noted that the decrease in rate of condensation from −10 F to +10 F outside temperature is greater for the roof over the attic than it is for the wall. This is due to the fact that for the attic the area through which the condensation leaves the attic is equal to the roof and gables, or about double the ceiling area, whereas in the wall the sheathing area is only equal to the inner surface area. Thus when the rate of vapor travel from the condensation area is increased, it has a greater effect on the rate of accumulation for the attic than it would have for the wall.

These test results demonstrate the theory shown in the calculations and indicate that the test conditions of constant outside air temperatures are more severe insofar as condensation troubles are concerned than equivalent variable air temperatures would be.

CONCLUSIONS

The conclusions reached from the investigation thus far made are as follows:

1. There are several fundamental causes for condensation within buildings, and in general they may be classified under three headings:
   a. Operating conditions for buildings.
   b. Construction of buildings.
   c. Outside weather conditions.

   In making an analysis of any problem all three should be considered, and in making corrections the causes under (*a*) and (*b*) must be coordinated to meet the requirements under (*c*).
2. The operating conditions which are usually subject to control are:
   a. Relative humidity carried within buildings.
   b. Controlled ventilation of structures.
3. The over-all vapor transmittance for a wall or combination of materials appears to be directly proportional to the vapor pressure drop across the wall providing that there is no condensation. The accumulation of moisture within any section of a wall is equal to the vapor transmittance from inside air to that section minus the vapor transmittance from that section to outside air. Reducing the relative humidity of the air within a building will reduce the rate of moisture condensation within the structure by a much greater percentage than the percentage of change in the humidity.
4. It is possible to reduce the rate of condensation within the structure by ventilating to the outside. This method may be particularly effective in attics where the condensation occurs on the underside of the roof. Adequate ventilation may be obtained without serious loss of heat.
5. Those factors which should be considered in new construction are:
   a. Build with special reference to vapor-proof construction on the inside of a building to prevent moisture from entering the walls or interior structure.
   b. Build with specific reference to allowing the vapor to vent through outside walls either with or without interchange of air.
   c. Use sufficient insulation to keep all interior surfaces above the dew-point temperature of the inside air.
6. A good vapor barrier on the warm side of insulation is an effective method of dealing with the condensation problem at the time of construction.
7. A barrier underneath the plaster will not interfere with the drying out of the wet plaster or affect the quality of the finished plaster.
8. A good vapor barrier construction may be obtained with a vapor-proof paper properly applied under the plaster or a vapor-proof finish on the interior surface of the wall.
9. For cold attic spaces it is desirable to allow openings for outside air circulation through attic space as a precaution against condensation on the underside of the roof even though barriers are used in the ceiling below.

ACKNOWLEDGMENTS

This paper gives a part of the results which have been obtained during the past year in a cooperative research program between the *National Mineral Wool Association* and the Engineering Experiment Station, University of Minnesota. The purpose of the program has been primarily to determine the cause of condensation within buildings, its probable extent under various conditions, and the best methods of preventing and correcting the same.

The authors wish to acknowledge the cooperation of the *National Mineral Wool Association* and the assistance of the Technical Committee of the Association in the entire program. This committee consisting of J. S. Irvine, *chairman;* W. W. Cullin, E. W. McMullen, E. R. Williams, and C. L. Neumeister has contributed liberally to the success of this project.

XUM

DISCUSSION

R. A. MILLER: I have been particularly interested in the discussion of moisture barriers which would effectively cut down condensation on the interior of buildings. In some of our work it has been necessary to develop means for eliminating the passage of water vapor and our studies have been in a very much lower order of magnitude than those with which Professor Rowley has been dealing.

In building a double-glazed unit, in which the condensation of moisture between the glasses may be a very serious factor in their permanent clarity, we have been relatively unable to find a barrier which we considered wholly efficient. When we talk in terms of asphalt and asphalt papers, from our standpoint such barriers are rather unsatisfactory.

In dealing with the total condensation on a house wall or a house ceiling, the possibility that the volume or amount of moisture under consideration may not be very serious probably would account for considering asphalt paper a satisfactory water vapor barrier.

A great deal of work has been done to arrive at something that will really prevent the penetration of water vapor, and little or nothing short of a metallic or a solid glassy, vitreous material appears to be particularly capable of definitely stopping some penetration.

The condensation of moisture on walls in the order of magnitude which Professor Rowley is discussing in the case of a double-glazed unit would render that unit seriously inoperative as a means of vision, and our figures, therefore, in dealing with such barriers are very much more exacting than the ones which Professor Rowley has discussed.

There seems to be nothing which will cut down vapor transmission to a degree which is at all comparable with a material called thiokol, one of the sulphur compounds, at any rate one of the plasticized sulphur materials which many of the rubber companies are experimenting with. Rubber itself does not appear to be an excellent water barrier from our standpoint. Apparently some metallic seal or some vitreous material to really prevent such condensation seems to be necessary.

I thought it would be of interest to the group to know that the study of moisture penetration through various materials has been given very deep thought by a number of different organizations and that asphalt paper and paper materials of that type do not offer the maximum values which may be obtained for that purpose.

PROF. F. B. ROWLEY: As Mr. Miller says, a vapor barrier is not 100 per cent efficient. It is not necessary to have 100 per cent barriers on the inside surfaces of most walls as there is always some opportunity for the vapor to continue through to the outside section. When moisture accumulates within a wall it means that the vapor is travelling through the inner section at a greater rate than it is travelling through the outer section. In order to eliminate the difficulty it is only necessary to reduce the rate at which it travels through the inner section until it is equal to or less than the rate at which it may travel through the outer section. Another point to be considered in the application of vapor barriers is that it would be practically impossible to get 100 per cent application even though the barrier was 100 per cent efficient.

PROF. W. T. MILLER: I am very much interested in this paper, especially in connection with an experience I had this winter. The Purdue University Research Foundation has what is called a test-tube village near the campus consisting of eight houses, all of different construction, and all of which have provisions made for

humidifying the interior air during the winter time. One of these houses is of 1-in. stucco construction, lath, 2 × 4 studding, and the inside finish consists of plywood panel. The wall is insulated with patent insulation, each batch being enclosed in a seal formed by means of a fire-stop across the 2 × 4 studding space. During the early part of winter and summer of 1938, the first year the house was used, sufficient moisture collected in the walls to run out on the floor and cause the floor to buckle and they could open only one outside door in the house. The fall of 1938 the same thing happened again, and an attempt to cure the situation was made. Incidentally, the plywood finish had commenced to deteriorate and rot and had to be replaced. When they removed the panel to replace the plywood they found the insulation was a nice sponge, full of water.

In order to study a method of curing this situation, we cut two 1-in. holes at the top of each of those insulation seals. At the sill plate of the studding space we installed needles to give us moisture data in connection with a moisture meter. We opened these holes in each insulation space in one studding space; in another one we put 1-in. holes at the top of a studding space only and connected the seals by boring three 1-in. holes through the fire-stops, and also had the moisture indication needles in the sill plate. Another we left the way it was.

We installed the moisture indicating needles in the sill plate and put the plywood panel back in place with screws so that it would be easier to remove for observation, and moisture readings have been taken at regular intervals since then. These holes were cut on November 21, and at that time the moisture content of the wood was so great we could not even estimate it with a moisture meter. It was so great that you could push water out of those 2 × 4's with your thumb. One week later the moisture content of the sill plate in the studding space that was vented at each insulation seal had decreased to 24 per cent. Incidentally, the maximum moisture that we could read with this meter was about 36 per cent. We do not know how much greater than 36 it was at the start, but we know that it was over 36 per cent. In one week's time that had been reduced to 24 per cent, and on December 21, two weeks later, it had been reduced to 18 per cent. On January 21 it was 15 per cent.

In the studding space which was vented by two 1-in. holes, and these holes connected or inter-connected by three 1-in. holes through the fire-stops, the first week showed no decrease that we could read. The second week it had reduced to 26 per cent and on January 21 it was 22 per cent. I am speaking of the moisture content of the wood in the sill.

On the panel that was not vented the meter tried to knock the pin off when we connected it across, showing that no moisture had been apparently escaping from the woodwork in the wall. While all of this was going on, a relative humidity of between 45 and 50 per cent was maintained in the house with a dry-bulb of 74. I have no data on the outdoor temperature vapor pressures, but the first part of our observation was taken during rather cool weather, in fact the coldest weather we have had at Lafayette, Ind., in the early part of November and December. It is interesting to me because the venting of the wall not only assists in eliminating the condensation of moisture but also permits the evaporation and escape of free moisture that is already in the wall.

PROFESSOR ROWLEY: I would like to emphasize the fact that the relative humidity of air within a building is one of the most important factors in many condensation problems. One might think that the rate of condensation within any part of a wall would be directly proportional to the relative humidity carried within the building, but this is not necessarily the case. When condensation takes place at some plane within a wall such as the inner surface of sheathing the maximum vapor pressure for this plane in the wall is determined by its temperature. The amount of moisture collected over a period of time represents the difference between rate of moisture

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travel from the inside of the wall to this section, and the rate from this section to the outside of the wall. Since the rate of travel through the inside portion of the wall is substantially proportional to the difference in vapor pressure of the air inside of the building and the vapor pressure corresponding to the temperatures of the material in the wall where condensation takes place, it is evident that the rate of vapor travel through the inner section of the wall will be in direct proportion to the relative humidity carried within the building. So long as condensation is taking place the rate of vapor travel from the plane of condensation to the outside of the wall will be independent of the relative humidity within the building. It will be governed by the temperature of the material at plane of condensation and the outside vapor pressure. Since the rate of vapor travel through the outer part of the wall will be constant a reduction of 50 per cent in the rate of travel through the inner part of the wall will reduce the accumulation by much more than 50 per cent. In checking up several cases of condensation in Minneapolis we have found relative humidities as high as 50 or 60 per cent during cold weather periods. Under these conditions one is almost certain to have frost formation within the walls, unless special precautions have been taken to use vapor barriers on the warm side. In most cases it is possible to entirely clear up condensation troubles by reducing the relative humidities carried within a building.

Professor Miller brings out two interesting points in his discussion; first the possibility of clearing up moisture within a wall by ventilating it to the outside, and second the conditions which may result from placing a vapor barrier on the cold side of a wall. Our ventilation tests have not been completed and therefore the results have not been included in this paper. I believe that the paper used on the cold side of the insulation for the house, referred to by Professor Miller, is a rather good vapor barrier; therefore when this material is placed on the outside of a wall it prevents any vapor which may pass through the interior section from passing on to the outside. This will result in condensation regardless of whether or not insulation is used. In the past the moisture problem within a wall has usually been considered as that caused by free moisture entering directly from the outside. To prevent this and also to stop air leakage building papers have usually been placed near to the outside surface. Recently many of these building papers have been so improved that they are good vapor barriers and therefore have a tendency to prevent any normal ventilation of vapor to the outside. The moisture problem has thus been aggravated.

In addition to the use of building papers on the cold side of walls which are often good vapor barriers, the normal relative humidities carried within buildings have been increased. This general increase has been partly due to artificial humidification and partly to the use of weather strips and other improvements which reduce the normal filtration of air between the inside and outside of a building. Usually an analysis of condensation problems will show the cause, and in most cases serious troubles can be eliminated by proper construction and reasonable operating conditions.

No. 1116

RESPONSIBILITY OF THE ASHVE TOWARDS THE SOLID FUEL INDUSTRY

By W. A. DANIELSON* (*MEMBER*), FORT KNOX, KY.

HEATING, ventilating, and air conditioning are derived almost entirely from fuel and in most cases from solid fuels. In the last two activities the fuel appears as power and the mechanical engineer with his long intense study has brought the combustion efficiency and incident equipment to a high standard of development. Heating, which is almost entirely in small plants using solid fuels, has had no corresponding study with the result that the greater portion of the fuel—soft coal—is rightly considered dirty, uncontrollable, and that it requires almost the frequent attention of a dusky slave.

This Society as the outstanding professional organization devoted to heating has, therefore, an obligation to the users of solid fuels. If it is to fulfill its mission it must ascertain the requirements of solid fuels for effective and convenient use and control and insure that this information is disseminated widely. That this is more difficult and involved than with gas or oil makes the need for competent investigation the more important. The reputation of the Society's impartial research activities will insure the acceptance of the findings by everyone.

Research in collaboration with the Anthracite Industries Laboratory at Primos has been initiated by this Technical Advisory Committee on a subject that has heretofore been at best an approximation. No known authentic data exist for chimneys for the average house. Certain rules of thumb now control design. Construction details will no doubt be much improved with a probable reduction in cost. This is given as an example of the work ahead in this broad field of study.

Anthracite coal has had size standardization for a number of years but that is not the case with bituminous. There is an optimum size for each service. To determine this requires the collection of much information and the Society's experience makes it the logical agency to analyze and reach definite conclusions.

Considerable information has been and is being published on solid fuels and while there are existing bibliographies on the subject more or less complete, those securing such information seldom know where to obtain guidance. The

* Chairman, Research Technical Advisory Committee on Solid Fuels.

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Society has a responsibility to not only its members but others seeking information on subjects relating to fuels. To disseminate this information the Technical Advisory Committee has created a sub-committee to prepare a bibliography of bibliographies. Arrangements have been made for this to appear from time to time in the Society's JOURNAL.

The simplest household fuel burning equipment should meet certain conditions, otherwise there may be danger to life and much higher cost than is necessary. The collection of information and cooperating with other research agencies where necessary can best be done by this Society; in fact there is no other recognized agency properly organized for this. Based on such study, codes should be prepared and it is in anticipation of these important guides for industry that three sub-committees have been appointed. Trade organizations have the right to expect this technical help and guidance.

Mining, transporting, and preparing solid fuels give employment to large portions of the population. In the constant reduction in work to be done, encouragement to continue as much as possible the activities which use labor is an obligation to general society. The Federal Government has definitely assumed leadership by recent legislation and the resulting establishment of the National Bituminous Coal Commission. This country-wide undertaking, reaching as it does into all industry and almost every individual's life, will require all the impartial and sound technical information that can be secured. Again, no other agency is in such a splendid position to be of real service in this undertaking as is the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

The Technical Advisory Committee, in assuming the work for the Society, tried to cover the entire field of the study and the methods by which it could carry its work into effect. It has adopted the program outline as given.

I. Scope and Purpose: The scope of the Committee's work shall be a study of the utilization of solid fuels with the ultimate objective of providing better service to the residential and small commercial consumer. The Committee is to collect and correlate facts from recognized authorities, cooperative with other agencies doing work in this field, and disseminate knowledge.

II. Information to be Collected:

A. Basic information on solid fuels.
1. Anthracite.
2. Bituminous coal.
3. Sub-bituminous coal, lignite and peat.
4. Coke.
5. Wood and charcoal.

B. Basic information on equipment.
1. Fireplaces.
2. Stoves and space heaters.
3. Boilers (low-pressure heating).
4. Warm-air furnaces.
5. Water heaters (residential, apartment, hotel).
6. Cooking ranges (residential, commercial).

C. Preparation of fuel for the market.
1. Sizing.
2. Cleaning.
3. Treatment.

D. Fuel delivery.
1. To the premises.
2. From bin to fire.

E. Combustion.
F. Removal of ashes.
  1. Removal of ashes from fuel bed.
  2. Removal of ashes from basement.
  3. Removal of soot and fly ash.
G. Requirements of the heat-absorbing surfaces.
  1. Heat absorption and transfer.
    a. Baffles.
  2. Ability to resist corrosion.
  3. Ability to withstand temperature changes.
H. Consumer satisfaction.
  1. Comfort.
    a. Capacity and rating.
  2. Safety and health.
  3. Convenience.
    a. Attendance.
    b. Control.
  4. Economy.
    a. Efficiency.
  5. Cleanliness.
  6. Dependability.
  7. Noise.

III. Sources and Methods of Collecting Information:

A. Collecting data from various organizations (published and unpublished).
  1. Government agencies.
  2. Trade association laboratories.
  3. ASHVE Research Laboratory.
  4. Universities and other educational institutions.
  5. Manufacturers.
  6. Fuel producers.
  7. Professional engineers and chemists.
  8. Libraries.
B. Papers written for discussion on the utilization of solid fuels.
C. Cooperating on research to find facts not already available.
  1. With governmental agencies.
  2. With trade association laboratories.
  3. With the ASHVE Research Laboratory.
  4. With universities and other educational institutions.
  5. With equipment manufacturers.
  6. In the field under the direction of the Committee on Solid Fuels.

IV. Dissemination of Knowledge:

A. To schools and colleges.
B. To the trade associations of the solid-fuel industries.
  1. Through reports submitted by this committee.
  2. By talks given at meetings of these trade associations.
C. To the trade associations of the heating industries.
  1. Through reports submitted by this Committee.
  2. By talks given at meetings of these trade associations.
D. To the Designers (ASHVE members, consulting engineers and architects).
  1. By papers written on the utilization of solid fuels.
  2. Through the ASHVE GUIDE.
  3. Through the manufacturers' technical staffs and literature.
  4. Through professional journals.
E. To the Manufacturers (executives, research staffs, salesmen).
  1. Through personal contacts.
  2. By correspondence.
  3. By articles published in newspapers and magazines written so as to be read by these groups.
  4. Through trade associations of the heating industries.

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F. To the Heating Contractors, Equipment Dealers, and Service Men.
1. Through manufacturers' technical staffs and literature.
2. Through ASHVE GUIDE.
3. Through trade associations of the heating industries.
4. Through trade magazines.

G. To the Solid-fuel Dealers.
1. Information distributed by trade associations of the solid-fuel industries and their representatives.
2. By articles published in trade magazines of the solid fuel industries.

H. To the Builders.
1. Through heating contractors and equipment dealers.
2. Through the designers.
3. Through solid-fuel dealers.
4. Through articles published in trade magazines.

I. To the Bankers, Real Estate Owners, Operators, and Advisors.
1. By articles published in newspapers and magazines.
2. Through heating contractors and equipment dealers.
3. Through the designers.

J. To Governmental Agencies.
1. Bureau of Mines.
2. Department of Agriculture.
3. Federal Housing Administration.

K. To the Consumer.
1. By non-technical articles published in newspapers and magazines.
2. Through solid-fuel dealers.
3. Through heating contractors and equipment dealers.
4. Through the designers and builders.
5. Through bankers, real estate owners, operators, and advisors.
6. Through manufacturers' advertising in magazines and newspapers.
7. Through educational institutions.
8. By exhibits, permanent and temporary, national and local.
   a. Heating and ventilating and similar expositions.
   b. Exhibits sponsored by solid-fuel dealers and trade associations.
9. By manuals on operation and service.

L. Through standards when information is definitely established.
1. Specifications, drawn up by the Committee on Solid Fuels.
2. Codes, by ASHVE code committees in conformity to requirements of *American Standards Association.*
3. Legislation, especially municipal construction codes, etc.

V. Committee Organization and Duties:

A. Advisory committee.
1. Outlining programs for working committees.
2. Voting on reports of working committees.
3. Dissemination of information.

B. Sub-committee on anthracite.
1. Collect and correlate information on anthracite and anthracite-burning equipment.
   a. Sub-committees on types of equpiment.
   b. Sub-committees to determine methods of comparing performance of fuels and equipment under the various items of *consumer satisfaction.*
2. Develop and propose minimum performance specifications.
3. Detailed cooperation with other research agencies.

C. Sub-committee on bituminous coal and coke.
1. Collect and correlate information on bituminous coal, coke, and equipment for burning same.
   a. Sub-committees on types of equipment.
   b. Sub-committees to determine methods of comparing performance of fuels and equipment under the various items of *consumer satisfaction.*
2. Develop and propose minimum performance specifications.
3. Detailed cooperation with other research agencies.

D. Sub-committees on other fuels.

No. 1117

SELECTION OF SOLID FUELS FROM THE VIEWPOINT OF THE SMALL CONSUMER *

By P. NICHOLLS,** (*MEMBER*), PITTSBURGH, PA.

THIS discussion will be restricted to a consideration of some of the factors associated with the selection of solid fuels and does not include the more general, and controversial, subject of comparison of solid, liquid, and gaseous fuels. Even with this restriction there is commercial rivalry in the claims advanced for coals and cokes.

Domestic appliances in which solid fuels are burned can be listed as hand-fired house heaters, stoker-fired house heaters, heater stoves, cooking stoves, water heaters and open fireplaces.

Each class of appliance has its particular requirements and requires some special treatment. Although the use of stokers is increasing rapidly, by far the greatest proportion of domestic fuel is still burned either in hand-fired house heaters or stoves, so those uses of solid fuels will be considered first.

For large plants the fuel may be selected on the basis of the true cost, that is, the cost either per unit of output of heat or power, and in industrial furnaces per unit of product. *True* costs include not only costs of the fuel, all direct labor and ash disposal, but also of such other factors as repairs, upkeep, and loss because of shut-downs. In most cases the estimation of the true costs are possible because money values can be assigned to most of the factors, and there are few intangibles to which values cannot be assigned. However, even for large plants, either opinions and preferences influence the weighting of factors, or a relatively small change in or addition to the equipment may alter the conclusions.

HAND-FIRED HOUSE HEATERS

In domestic heating the money value to the household, or the true cost, is more difficult to evaluate. A householder usually considers cost as limited to

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** Supervising fuel engineer, Bureau of Mines, Central Experiment Station.

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the direct expenses for fuel and to the removal of ashes from the premises. Normal repairs or replacements for a furnace do not occur frequently enough for him to relate them definitely to the fuel, unless they are due partly at least to his own lack of care. An individual householder cannot assign definite relative expenditures for cleaning the house or its furnishings as caused by his own use of a fuel, because he cannot allocate the blame between himself, his neighbors, or general atmospheric conditions.

Time and labor given to the care of the furnace are true expenditures, although for some rare individuals tending a furnace is a pastime. The relative attendance required by different fuels is most simply expressed by the frequency of attendance, or the time intervals between them, required to give the same results, that is, the same degree of comfort. Although this method of expressing attendance is satisfactory when fuels of the same type are compared, other factors, such as total time spent on the furnace and relative disagreeableness of the jobs are important when different fuels, such as coals and cokes, are compared. Moreover, it is not always true that the same degree of satisfactory heating can be obtained even though the attention to the furnace is unlimited; if a fuel requires more draft than is available in the equipment, satisfactory heating will not be possible. What usually happens is that one is content with a decrease in uniformity of comfort in order to save visits to the furnace.

ATTENDANCE FACTOR

In general, the attendance required decreases with increase in the price of fuels, and one is willing to pay a higher price because the furnace will require less attention. As an alternative, money can be invested in automatic equipment and regulators.

The different types of attention are (*a*) frequency of firing, (*b*) attention between firings, (*c*) shaking of grates and removal of clinkers, (*d*) cleaning of ashpit, and (*e*) cleaning furnace of soot and fly ash.

On the assumption that the same volume of fuel is fired, the interval between firings is proportional to the product of bulk density, calorific value, and over-all efficiency. On the basis of lasting quality of fuels, anthracite lasts longest, bituminous coals next, and cokes the shortest time. However, other factors affect the frequency of firing in practice. Few attendants fill the firepot to its capacity; usually they get the habit of firing either two or three times a day so that the fuel added at each firing will be only enough to last until the next firing. Moreover, few will, and no one should, add a heavy firing of bituminous coal at one time; rather they would build up the bed in stages, which increases the number of visits to the furnace. As an alternative, a larger weight can be fired by adding the labor to pile the hot fuel to one side of the box and fire the new coal in the hollow.

Coke and anthracite coal require no attention to their fuel bed between firings provided the bed is sufficiently cleared of ashes at the time of firing so that the available draft can maintain the rate of burning as more ash accumulates in the fuel bed. The number of attentions required by coals which cake will vary with the coal, the weather, and the skill of the attendant;

controls to regulate the temperature of the room or to limit the temperature of the water or air cannot function if the bed is caked into big lumps. Badly caking coals thus require the added attendance for breaking the lump of coke.

The relative time and trouble required to remove ashes from the bed depend partly on the quantity of the ash, but more on its clinkering characteristics, although it does not follow that an ash with the highest fusion temperature is the most desirable or that one with a very low fusion temperature cannot be used satisfactorily. The most desirable ash is one that will sinter into a mass that is porous to air but is friable. Such a formation is easily broken yet is not dusty. An ash that fuses little and gives a fine light dust is not only disagreeable to handle and gets through the house, but also interferes with the regulation of burning because of the high resistance to air flow that it adds to the bed.

Although decrease in rate of burning can be overcome by more frequent shakings, the average effect of fine ash is disadvantageous and increases the attendance required.

TABLE 1—MAXIMUM DROP IN PRESSURE THROUGH FUEL BED, INCHES OF WATER

RATE OF BURNING, POUNDS PER SQUARE FOOT PER HOUR	MEAN SIZE OF FUEL, IN.			
	¾	1¼	1¾	2½
1½	0.072	0.064	0.050	0.030
2½	0.099	0.091	0.092	0.113
3½	0.108	0.107	0.132	0.178

As is usual with most alternatives, one cannot have an ash that is best for all conditions; one that sinters at lower rates of burning will form slag and clinkers at high rates.

The behavior of ash as it may affect operation can be quite complex. Table 1 shows the maximum pressure drop through the bed at three rates of burning of various sizes of coke having an ash-softening temperature of 2500 F. At the lowest rate the maximum pressure drop decreased materially with increase in size of coke; at the medium rate there was a small increase in pressure drop with increase in size; and at the highest rate the increase was large. These results are due to the relative amount of fusion of the ash under each set of conditions.

The quantity of clinker and its nature depend not only on the fusion temperatures of the ash but also on the form of the ash in the fuel, that is, the quantity and fusibility of the bone and slate. It is desirable that all clinker shall pass to the ashpit and that none need be removed through the firing door; whether or not the latter is necessary depends not only upon the type of clinker but also upon the kind of shaking grates in the furnace. The most fusible ash is not necessarily the most troublesome; a small piece of hard rock or clinker that fouls the grates may necessitate more labor in removing it than will a larger block of clinker. For example, the ash of a coke made from a southern Illinois coal was so fusible that all of it melted and formed one

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mass of clinker. However, the mass was of the honeycomb type and was porous; consequently, it could be allowed to build up for several firings and was then easily removable in one piece, involving less total labor than do coals with more refractory ashes.

There are no hard and fast rules for evaluating the relative labor factors because they will vary with the furnace and its operation; if the furnace is small for the house or if the method of regulation is from full open to complete closing of the dampers, there will be more clinkering than in equipment in which the rate of burning is more uniform.

With coking coals the clinkering does not depend much upon the size of the pieces of coal, but with free-burning fuels the clinkering will, in general, be more severe as the size of the fuel is smaller, because the maximum temperature in a fuel bed increases with a decrease in size. For this same reason of higher fuel-bed temperatures the clinkering in a warm-air furnace will be greater than in a boiler of equivalent heating capacity, because the water-cooled firepot reduces the temperature of the bed.

In cleaning the ashpit the preference is for an ash that fuses, both because it decreases the fly dust and because it occupies less volume.

The frequency with which the flues of a furnace must be cleaned and the disagreeableness of the job are related directly to the smoking and sooting characteristics of the fuel and, from all aspects, a smokeless fuel is desirable. In few households will the furnace be kept clean enough so that the accumulations of soot in the flues and stack will not decrease the available draft and thus hamper the best performance of the furnace.

RETURNS AS HEATING SATISFACTION

The returns obtained for the expenditures per ton of fuel plus the labor and time of attendance involve the satisfactoriness of the heating system as a whole, but the comparison of fuels for a given house can be based on (*a*) the heat delivered per ton of fuel and (*b*) the satisfactoriness of its regulation.

The relative heat delivered by different fuels will be closely proportional to the product of the calorific value and over-all efficiency. Bureau of Mines Bulletin 276 [1] contains the most comprehensive data for American coals burned in hand-fired domestic boilers and reports the results of more than 500 tests using 184 different coals. The analyses of the results show how average efficiencies and heat delivered are related to such main characteristics of the coals as moisture, ash, fixed carbon, volatile matter, and size. Some of the conclusions for the average of these tests were:

1. The thermal efficiency of the fuel actually burned, that is, excluding the loss to the ash pit, increased gradually as the volatile decreased until the volatile was about 18 per cent, after which it increased more rapidly, in the case of the anthracites; also, the thermal efficiency on the average was independent of the ash content of the coals up to 15 per cent ash.

2. In these tests the higher thermal efficiency of the semi-bituminous coals and anthracites was offset by higher losses to the ash pit; moreover, the anthracites

[1] Five Hundred Tests of Various Coals on House-Heating Boilers, by P. Nicholls, S. B. Flagg and C. E. Augustine. (*Bulletin 276, Bureau of Mines*, 1928, 70 pages.)

at that time had a high ash content. As a result the average heat transferred to the water per pound of coal burned was approximately the same (about 7800 Btu) for all coals having below 35 per cent volatile matter on the ash- and moisture-free basis.

In another series of tests,[2] all made in the same domestic boiler, the average heat transfer to the water was as given under *A* in Table 2. Because the ash in the anthracites used was high, the results are modified to an assumed 11 per cent ash.

The values under *B* are the results of a limited number of tests made by the Battelle Memorial Institute[3] of heating plants, mainly warm-air furnaces, in houses. The relative costs for the fuel for a season's heating with these hand-fired furnaces were estimated to be: Bituminous coal $100, semi-bituminous $89, and coke $87. These costs include the price of the fuels and the cost of ash removal at Columbus, Ohio.

The lack of agreement among such tests shows the dependence of results upon conditions. Anthracite and coke have the advantage in that their efficiency

TABLE 2—HEAT TRANSFER RATES FOR VARIOUS FUELS

	FUEL	CALORIFIC VALUE	OVER-ALL EFFICIENCY	HEAT TRANSFERRED TO WATER PER POUND OF COAL, BTU
A	Bituminous coal	13,610	46.7	6,400
	Anthracite	13,000	64.0	8,340
	Coke	12,610	66.6	8,400
B	Bituminous coal	13,400	45.0	6,000
	Semi-bituminous	14,600	50.0	7,300
	Coke	12,350	70.0	8,600

is less affected by lack of skill in firing than is that of bituminous coal, the efficiency of which may be materially decreased by loss of volatile matter because charges are too heavy or the fire is smothered. The over-all efficiencies of bituminous coals in the tests reported in Bulletin 276 were high because the firings were usually at 6-hour intervals and because the methods of firing were such that the hot bed was not buried. On the other hand, in the tests of the semi-bituminous coals and anthracites the losses of combustible to the ashpit were high.

The relative degree of satisfactoriness and uniformity of heating with different fuels is dependent on the required attention being given to the furnace. Any fuel, within reasonable limits, will produce satisfactory results provided the furnace is given enough attention. The probability of this being done decreases as the time required is greater, thus free-burning fuels such as anthracite and cokes have the advantage; however, they lose some of this advantage when the installation includes thermostats or automatic regulators.

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[2] Heat Transference and Combustion Tests in Small Domestic Boiler, by H. W. Brooks, M. L. Orr, W. M. Myler, Jr., and C. A. Herbert. (ASHVE TRANSACTIONS, Vol. 31, 1925, p. 185.)

[3] Efficiencies and Costs of Various Fuels in Domestic Heating, by R. A. Sherman and R. C. Cross. (*Technical Report No. 3, Bituminous Coal Research, Inc.*)

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The foregoing generalities cover all kinds of fuels, but a householder has only a limited number from which he can make his selection, and the kinds available depend largely on his geographic location. In the States of the Atlantic seaboard the broad choice is among anthracites, low-volatile bituminous coals, and cokes, all in the smokeless class. Proceeding westward there are the medium-volatile bituminous coals in central Pennsylvania. The high-volatile coals extend to Indiana, with gradual lowering of rank, that is, with decreasing calorific value, and also with more fusible ash. In the Rocky Mountain province the coals mined range from anthracite to lignite, with large areas of sub-bituminous.

In general, householders do not change the kind of fuel they are using each year, but rather their problem is to select from the available sizes or grades of one kind. Most users have fairly definite preferences as to size, but their choice between grades is associated with trade names, or with certain retail distributors, rather than from a technical knowledge of differences in properties. Apart from the desirability of a certain size because of its satisfactoriness, householders prefer large sizes in the belief that small sizes contain more ash and have a lower heating value; this belief is justified except for coals that have been cleaned or for low-volatile coals, the carbonaceous matter of which is fragile and thus predominates in the fines.

Size is of prime importance in noncaking fuels, but there is some leeway in selection. For anthracites and cokes the choice is usually between stove and chestnut. Stove, the larger size, has the advantage that less secondary air is required and the temperature through the bed is more uniform. For the same degree of satisfaction the fire must be replenished sooner with stove than with nut. In general, the nut size is easier to handle, especially with hard cokes, because the fire is less apt to go out in mild weather. The preferable practice is to use the smaller size in spring and autumn and the larger when there is real winter. The severity of clinkering increases with decrease in size, but not so materially when stove is compared with nut as for still smaller sizes.

Size becomes of less importance as the caking quality of a coal increases. In earlier times, when few mines had any but the crudest preparation plants, householders received run-of-mine and learned how to handle the big lumps. Such big lumps could be used effectively in mild weather, and many still prefer them, but better uniformity of burning can be obtained with sized coals. A number of sizes are usually available and may range from 6-in. lumps down. No study has yet been published to show the advantages or fine distinctions of different sizes of coking coals; it may be expected that the most satisfactory size will depend on the caking characteristics and the method of firing and that the size should decrease with a decrease in the area of the fuel bed.

Although fuels under each classification have certain common burning characteristics the values of these characteristics vary, and thus a purchaser may have a range of choice. Some examples are given.

Anthracites have the same general burning characteristics, but the range of 2 to 8 per cent in volatile matter in the coals from mines in various districts is enough to make the differences in their burning noticeable. For the same size of pieces and with other conditions the same the coals with the higher

volatile will have a quicker pick-up after a firing or a banking period and also burn more easily. Against this the anthracites with low volatile are denser and a greater weight can be fired.

Coke to a householder means high-temperature cokes, which do not have more than 1½ per cent volatile matter. Cokes can be divided into two types, gas-house and by-product, but it is only in a few districts that both types are available. The physical hardness of cokes is a good measure of their differences in burning properties. A dense, hard coke that rings when dropped is less reactive than a soft coke, will be harder to ignite, and the fire more liable to go out in mild weather; on the other hand, it is denser and a greater weight can be gotten into the furnace. In general, gas-house cokes are softer than those of by-product origin. The ideal arrangement would be to purchase the softer coke for autumn and spring and the hard coke for winter, but, as explained previously, the same result can be partly accomplished by buying two sizes.

The range in qualities and burning characteristics of bituminous coals is so much greater than for anthracites or cokes that they should be considered under subdivisions and with reference to the geographical location of use. High-grade coals, such as the Pennsylvania and Appalachian, are shipped into districts where lower-grade coals are mined, but rarely is the reverse true. Thus in Pittsburgh, for example, the choice of kinds of bituminous coals covers only a narrow range, whereas in Columbus the range runs from lower-rank coals mined in Ohio to higher-rank imported coals. However, in all districts there is the choice among coals similar as far as the combustible part is concerned, but differing in ash content and ash fusibility. In the merchandising of coal the stage has not been reached where the delivery slip guarantees the quality of the coal by giving its calorific value, ash content, and ash fusibility; therefore, the purchaser must depend upon the coal dealer or the reliability associated with a trade name, mine, or producer.

Bituminous coals, even though they are of the same rank and heating value, may differ in their behavior in domestic furnaces, but as yet few systematic data have been obtained on the finer distinctions; the most complete data are those for the 184 coals in Bureau of Mines Bulletin 276, but they are for wide ranges of coal rather than for finer distinctions of coals from different mines in a district.

Some characteristics that a householder will notice are the nature of the caking or how the coal fuses into large masses; how this caking reduces the burning by shutting off the draft; the difficulty of breaking up the caked mass; the size and strength of the coke formed; the nature of the ash and clinker; and the quantity and clogging of the soot. As previously stated, the over-all measure of the desirability of coals is the nature and amount of attention required to give the same degree of satisfactory heating.

In the Western and Southern States, where sub-bituminous coals or lignites are available, additional characteristics are noticeable. Their high moisture and tendency to weather, disintegrate, and ignite in storage introduce special problems when these coals are used in the common type of hand-fired furnaces.

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In general, they are better fitted for a magazine feed which will maintain a thin fuel bed.

Figs. 1 and 2 are reproduced from Bureau of Mines Bulletin *276*. They show the relationship of the average efficiencies obtained and the properties of the coals. The numerals in circles give the number of tests that were averaged for each value. Fig. 1 gives the thermal efficiencies; the top curve is for the efficiency of the combustible burned, the next for the percentage loss to the

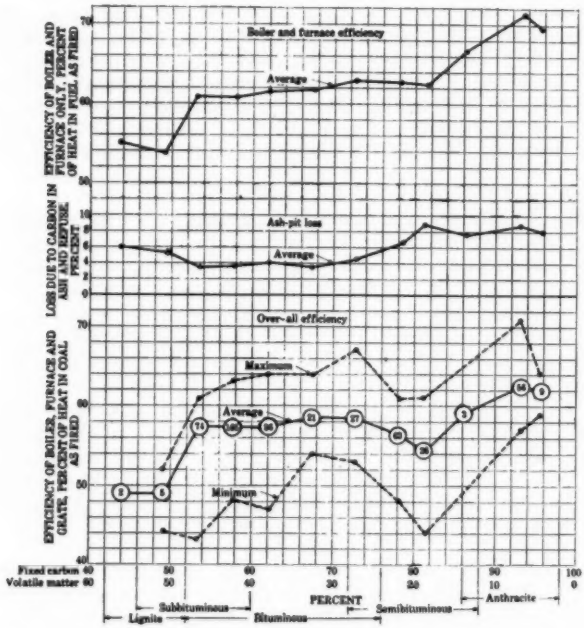

FIG. 1. EFFICIENCY, ASH-PIT LOSS, AND HEAT ACCOUNTED FOR WITH VARIOUS PERCENTAGES OF FIXED CARBON AND VOLATILE MATTER IN COAL, MOISTURE AND ASH FREE

ash pit, and the bottom curves for the over-all thermal efficiency. Fig. 2 shows the relative heating values per pound of coal as fired; the three upper curves show the moisture, ash, and calorific values of the coals. The figures illustrate the broad relationships of the results obtained, but do not show the relative troubles they caused; data on attentions are given in the bulletin.

FUELS FOR DOMESTIC STOKERS

Although there are several types of stokers, differing in principle and in the fuel best suited to them, the great majority now are of the underfeed type

with screw feed. From the standpoint of the manufacturers it is desirable that one style of stoker should be suitable for a wide range of coals. At present there is a general division of stokers into those for anthracites and those for bituminous coals; it may be expected that there will be a tendency

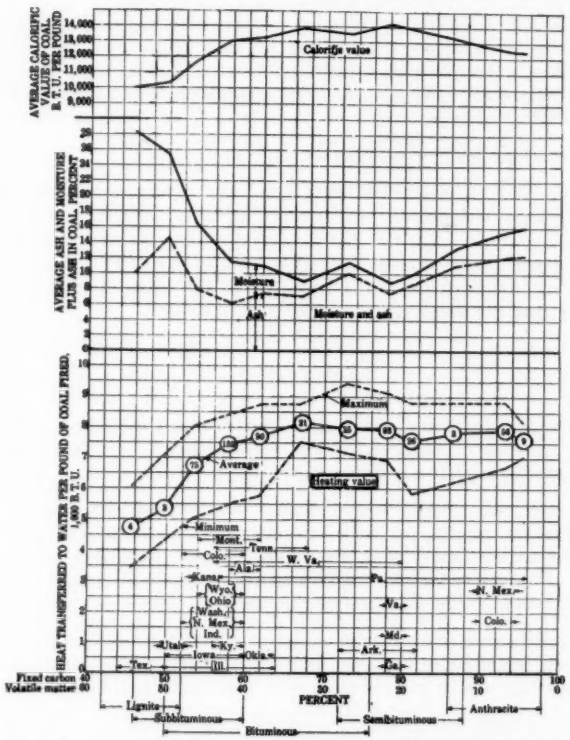

FIG. 2. RELATIVE HEATING VALUES OF COALS WITH CORRESPONDING CALORIFIC VALUES AND PER CENT OF MOISTURE AND ASH

to subdivide the latter class so that, even though the modifications are small, stokers will be available to give the best results for a narrow range of coals.

Stokers are of recent enough development that the householder has no experience upon which to base the selection of his coal but has to depend on recommendations he receives, first from the dealer who sells him the stoker and afterwards from coal salesmen.

Manufacturers of stokers for anthracite have the advantage of knowing definitely the range of variation of the coals that may be used; tests on

anthracite stokers have subordinated the dependence of results on the type of anthracite, that is on the volatile content, and have emphasized the effect of size.[4] Sizes from No. 2 buckwheat to pea have been used; No. 1 buckwheat is generally recommended, but pea has advantages with larger stokers.

Full knowledge of the relative desirability of bituminous coals for underfeed stokers as sold is limited, but much useful investigative work has been done. It has been shown that most coals can be used with some success but that as a general rule the desirable qualities[5] are: (1) not more than moderate caking properties; (2) an ash-softening temperature of 2100 to 2550 F; and (3) a size of 1¼ to ¾ in.

These limitations are not definite but only preferable.

Most furnaces with magazine feeds, solid-fuel hotwater heaters, and other burners have been designed for and tested with definite fuels, and householders having such apparatus have no choice except to obtain the recommended fuel at the lowest price. Successful operation of such furnaces depends upon the correct flow of the fuel and on it being of a size suited to the thickness of the fuel bed that the apparatus is designed to produce.

CONCLUSION

The most important decision a householder is asked to make is whether the elimination of smoke is desired. Having done this, it should be possible to rely on the guidance of the retailers in details of selection.

Until recently coal producers and dealers did not consider that they should act as combustion experts for their customers, who had to depend upon their own common sense when buying fuel and had little published information to help them. Now many of the larger coal companies and some retailers have trained combustion engineers to guide them and help their customers.

Of equal importance has been the action of associations of the fuel industries in establishing laboratories to study their fuels and to desseminate information to insure that they will be so used as to give better satisfaction to customers.

There are separate laboratories and associations for the gas industry, the oil burner industry, the anthracite industry, and the bituminous-coal industry. In addition there are many smaller associations of producers and retailers. These can best help their indlustries by guiding householders to select fuels so as to obtain the most satisfaction.

DISCUSSION

A. W. MARSHALL: I have no criticism of this paper, but I would like to expand on one thought, and that is relativity in the selection of fuels. What suits one person does not suit another. One person is looking for the very lowest cost; perhaps

[4] Domestic Anthracite Stokers, by P. A. Mulcey. (*Transactions* of First Annual Anthracite Conference, pp. 83-93.)
[5] The Preparation of Stoker Coal, by R. A. Sherman. (*Coal Heat*, Vol. 34, 1938, pp. 5-8.)

more of them are looking for comfort and convenience, low ash, ease of clinker removal and even the odor of clinkers.

In the Chicago district we have many high sulphur coals. In residential fuel, the greatest impediment to the high sulphur coals is the odor of the clinker in occupied buildings. That has been no small factor in causing this $25,000,000 to be spent in the two states in the preparation and particularly the washing of coal to lower the ash, but also to lower the sulphur. As Mr. Nicholls says, the house with the twelve-year-old boy has no problem of ash removal and shoveling of coal. A home where the man is away for periods of time selects low ash coals, so that there are many factors outside of efficiency and outside of ultimate heating cost that have to do with the selection, so that it is almost impossible to formulate any rule.

In making a talk before the Illinois Mining Institute, the producers asked me to bring this out; while there has been $25,000,000 spent on the physical side of the mines, it will be necessary to spend many more millions of dollars in the reconstruction of the entire sales organization of the retailers, which means that better trained men for the producers to educate retailers who can fit the coal to the unit are going to be necessary. By *fitting* I mean such an ability as not to give too high a fusion coal for this plant and there is such a thing as too high a fusion coal, or not to give too low a fusion coal to that plant, but to fit the coal to the home or the plant physically and psychologically. It will be a tremendous job and even a harder job than the spending of the $25,000,000 to prepare the coal.

H. J. ROSE: It is gratifying that Mr. Nicholls emphasized heating satisfaction in his paper, because the solid fuel people are realizing more and more the importance of that feature. The criticism has often been made of stove and boiler manufacturers that they think primarily of such equipment as so much cast iron. The criticism has likewise been made that combustion engineers test chiefly for combustion efficiency, which is certainly not the most important factor to the domestic user.

The new Technical Advisory Committee on Solid Fuels has emphasized the importance of consumer satisfaction. The consumer's first requirement is comfort, that is, adequate and controlled heat. Second in importance is convenience, the freedom from frequent attention and from dirty and disagreeable work. Safety and health factors are also basic. While economy is important, it is not the consumer's first requirement except in the lowest income class. Cleanliness, dependability, and freedom from noise in the case of mechanical equipment are other essentials for consumer satisfaction.

With these thoughts in mind, the Committee on Solid Fuels will collect and develop methods for evaluating such factors. Eventually codes or other appropriate specifications can be established which will set forth minimum performance standards which domestic equipment for solid fuels must meet in order to insure consumer satisfaction and to meet the competition from other fuels which are used automatically.

J. D. DOHERTY:[6] I would like to question the high ash-pit loss indicated for low-volatile coal such as Pocahontas. As I recall it, the percentage of fines in the low-volatile coals used in these tests was abnormally high and in all probability the firing method employed was not that of a person accustomed to this type of fuel. Even with excessive fines a high ash-pit loss should not be expected with coal such as Pocahontas (except in tests of short duration) as the coking characteristics of the coal convert fines to larger masses when they are added to a going fire.

Our tests indicate a very low ash-pit loss for this type of coal and examination of results that have been reported by the Department of Mines, Canada, Bituminous

[6] Koppers Coal Co., Pittsburgh, Pa.

Coal Research Inc., and in the TRANSACTIONS of ASHVE will, I believe, make this point clear.

E. K. CAMPBELL: In regard to Table 1, with the low rate of combustion, a larger size fuel showed less resistance through the fuel bed; with a higher rate of combustion it shows greater resistance through the fuel bed. Can Mr. Nicholls give me an explanation of that?

PROF. A. P. KRATZ: I would like to point out that our experience in the research residence checks with Mr. Doherty. Our ash-pit loss either with bituminous or with anthracite coal never exceeded 3 per cent, and I do not think we used any extraordinary methods of firing.

P. NICHOLLS: Two speakers commented on the high ash-pit loss shown by Fig. 1 for the semi-bituminous coals. There is no error in the relative values of these losses as compared with the other coals, and they are the averages for over 90 tests. Data given in the Bureau of Mines Bulletin 276 show that most of the semi-bituminous coals had excessive fines. These tests were made for other government departments and their object was to determine what the relative results would be when commercial furnaces were fired by the average man, and not what could be done with extra skill and care; both the alternate side-coking and the center-cone methods of firing were used with the semi-bituminous coals. The firemen were ex-army men who were representative of those likely to have the care of heating boilers in small government buildings.

Mr. Campbell's question on Table 1 is answered in the text. These data on pressure drop were given to show the unexpected relationships which occur because of the different forms of the ash and clinker. Except for small sized fuel, most of the drop in pressure through the fuel bed is not caused by the fuel but by the accumulation of ash. If the ash is dust the pressure drop will be high; thus a coal with a high-fusion ash will require more draft at low rates of burning than one with low-fusion ash, because the ash of the latter will cinder. At high rates, coal with the low-fusion ash may require more draft because now its ash may fuse and form a sheet clinker, whereas the high-fusion ash will only cinder. In the tests of Table 1 the temperature through the fuel bed decreased as the size of the pieces increased; thus the ash was less fused with the larger size fuel and there was more as dust. However, it does not always follow that a fuel with a low-fusion ash will cause a lot of trouble. We tested, and I also burned in my home, a low-temperature coke having an ash with the lowest fusion temperature of any we have had, and yet it was not troublesome because it formed a porous clinker which let the air through. In my own furnace I would allow the clinker to accumulate for two or three days, and then could remove it in one piece. As all the ash fused and there was very little dust, there was less trouble than with a high-fusion ash before removing the clinker but the fire should be allowed to burn rather low so that the clinker is not too hot or the fumes from it may be disagreeable.

No. 1118

SMALL STOKERS

By PAUL A. MULCEY,* PRIMOS, PA., AND RALPH A. SHERMAN,**
COLUMBUS, OHIO (*MEMBERS*)

Part I. Domestic Anthracite Burners

ONE of the few industries which has shown a definite upward trend during the past eight years is that of domestic anthracite burner production. Starting from very humble beginnings, there were only a few large companies manufacturing stokers prior to 1930, and the number of units sold was small. In 1936 anthracite stoker sales amounted to 5,000 units. This figure was doubled in 1937, and is expected to be tripled in 1938. For the first eleven months of 1938 the industry was 42 per cent ahead of the same period in 1937.

Accompanying the accelerating burner output, there has been a definite trend toward lower prices. As a matter of fact, the domestic anthracite stoker is now in a position to compete with other automatic fuel burners. The probabilities are that as time goes on more and more anthracite stokers of increasingly simplified design, and hence lower first cost, will gradually find usage in those territories which up to the present time have been considered as the exclusive market of competitive fuels.

The purpose of this paper is to discuss the general mechanical aspects of small anthracite stokers with particular emphasis on those parts of the mechanism which are peculiar to the burning of hard coal; likewise, to attempt to draw a picture of the present state of development of stokers.[1, 2, 3, 4]

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* Assistant Director, Anthracite Industries Laboratory, Primos, Del. Co., Pa., Author, Part I.
** Supervisor, Fuels Research Laboratories, Bituminous Coal Research, Inc., and of Fuels Division, Battelle Memorial Institute, Columbus, Ohio, Author, Part II.

[1] Domestic Burners for Pennsylvania Anthracite. (*National Bureau of Standards*, CS48-34, 1934.)

[2] Performance Expectancy of Domestic Underfeed Stokers for Anthracite, by Allen J. Johnson. (Technical Publication No. 608, *AIM & ME*, 1935.)

[3] Small Underfeed Single Retort Stokers, by B. M. Guthrie. (*Transactions ASME*, 1938, Vol. 60, No. 1.)

[4] Domestic Anthracite Stokers, by Paul A. Mulcey. (*Transactions* First Annual Anthracite Conference of Lehigh University, 1938.)

Presented at the 45th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Pittsburgh, Pa., January, 1939.

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GENERAL PRINCIPLES OF DESIGN

The domestic anthracite stoker may be divided into five main component parts. These are:

1. Coal conveyor system.
2. Ash conveyor system (omitted in certain types of stokers).
3. Fan and air distribution system.
4. Combustion retort, at which point the foregoing converge.
5. Drive mechanism.

Fig. 1 shows in cross-section how these five parts are correlated.

Items 1 and 4 are those which present the greatest problems to designers, and which will be considered in some detail in this paper. The ash conveyor, although an important part of many anthracite stokers, does not present any

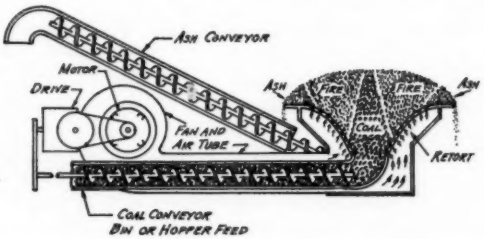

FIG. 1. COMPONENT PARTS OF TYPICAL ANTHRACITE BURNER

important obstacle, nor is its design particularly critical. The problems involved are those of materials handling, mechanical layout, and noise elimination. The removal of anthracite ashes, either by gravity to pits or mechanically, is fortunately one of the minor items in stoker design.

The supply of air in an anthracite stoker is a problem which may be treated from a general knowledge of the subject of fans and ducts. No special volume regulators or pressure compensators are required, since combustion in a well designed anthracite retort is uniform. Under the subject of retorts the problem of air distribution in the fire will be discussed a little more fully.

Many types of drive are used in anthracite burners. However, the essential elements consist of a motor which turns the fan impeller either directly or through pulleys and at the same time moves the coal and ash conveyors by means of a speed reduction element. The latter may be of the gear or hydraulic type. Under normal conditions the fan consumes the greatest portion of energy supplied to the stoker unit as the remainder of the mechanism requires only a small amount of power.

Detailed design of the retort will be considered later. However, mention of the mechanical means sometimes used to expedite the removal of ash from the

retort may be necessary. Mechanical *agitation* is generally divided into two types: (*a*) that caused by a vertical scraper arm-rotating about the retort or other moving pieces serving the same purpose; and (*b*) that caused by rotation of the retort itself. Obviously, mechanical agitation permits the use of smaller clearance between the retort and side walls of the furnace than is required in the *spillover* type of stoker in which the ash falls off by its own weight.

While the drive is an extremely important part of the stoker, it need not necessarily be different in design and layout for an anthracite burner than for a soft coal stoker.

TYPES OF ANTHRACITE STOKERS AVAILABLE

In order to indicate the relation between the retort and coal conveyor, Fig. 2 shows schematically the four types of bin feed stokers and the hopper type. The latter is rapidly being supplanted by the more convenient bin feed stoker which completely eliminates the manual handling of coal. The first unit shown is the straight line burner with a single conveyor feeding directly from the bin. This type ordinarily has its drive unit and fan on the opposite side of the heater from the bin, although in the case of a plunger type stoker it may be located on the same side. The second burner is also a single conveyor, straight line machine, but the coal conveyor is driven through a shaft and sprocket from the bin end of the conveyor. The third is the straight line, intermediate drive type with the drive mechanism located between the two conveyors. To balance the slight complication of two conveyors, this unit is somewhat more compact. This stoker can also be considered a special case of the *universal* type. The latter stoker permits the bringing of coal from a bin located at any point within a reasonable distance from the furnace, since the conveyor joining the bin and mechanism may be swung at any angle. Finally, the hopper type stoker is, of course, the simplest and is in effect a small version of the straight line bin feed of type 2.

ANTHRACITE CONVEYORS

In the design of conveyors to transfer anthracite from a hopper or bin to the burning retort, the factors to be considered come under the two general headings of mechanical and chemical (corrosion resistance) considerations. The system must be so designed as to eliminate excessive congestion and at the same time materials must be selected which will withstand the corrosive action of vapors which may be released at certain points in the conveyor. In addition to this, every effort must be made to minimize the concentration of such vapors.

Details of design have been specified by a number of manufacturers. Long trial periods were necessary before any stoker was definitely considered satisfactory. Obviously, therefore, a rather wide variation in coal conveyor design is found both from the mechanical and materials standpoint. For example,

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conveyor tubes vary in inside diameter from 2 to 4 in. while being constructed of such divergent materials as wrought iron and brass.

A systematic survey of coal conveyors will be made by starting at the bin end of such a unit and following it through to the retort. The inlet end, particularly in the case of models feeding from the bin, is very often so designed as to feed coal into the tube at such a rate as to preclude the possibility of loading the conveyor to capacity. The principal methods of accomplishing this are by having a length of screw extending from the conveyor tube into the bin and having, (*a*) a smaller diameter; (*b*) decreased pitch; (*c*) larger shaft diameter; or (*d*) intermittent screw flights. Another method is to so modify the entrance to the tube as to restrict the flow. The latter has the disadvantage of causing increased breakage of the coal, although by proper

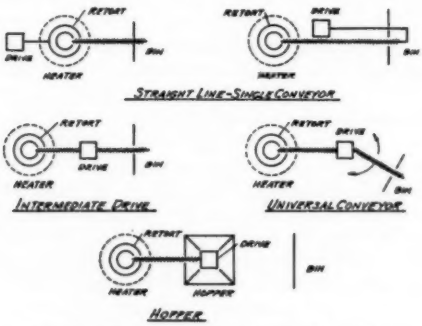

FIG. 2. VARIOUS METHODS OF COAL FEED

design this may be offset by the advantage of preventing the entrance of oversize coal or foreign material which might jam the mechanism. As contrasted with bituminous stoker practice, the use of *crushers* at the intake is to be avoided. American anthracite is so prepared that each size marketed varies within well-defined limits. The small amount of accidental oversize encountered is not enough to warrant the use of mechanisms which by their very nature increase breakage and reduce combustion efficiency. On the other hand, the design of bin conveyors to provide readily accessible cleanouts for the removal of large coal and foreign material is highly recommended.

In feeding coal from a bin a desirable feature is to have a relatively large *active* area which will be served by the conveyor. In other words, the dead space from which coal must be moved manually to the conveyor must not be excessive. For this reason many bin feed stokers have relatively long open screws, either with a bearing at the end or floating in the coal. One conveyor of the latter type is made to bring coal in from a fairly large area by extending the screw flights about 40 in. beyond the end of the shaft, thus providing a relatively flexible feeder.

Manufacturers often recommend that bins be constructed with a *hopper* bottom in order that all of the coal will feed by gravity to the conveyor inlet.

However, the author believes that a certain amount of dead storage volume is quite desirable to take care of emergencies arising when the coal bin is supposedly empty and a fresh supply of fuel is not readily available.

About 50 per cent of anthracite stoker conveyors have a theoretical feeding capacity equal to that of the portion of the conveyor extending into the bin. The remainder increase in theoretical capacity over a range of 36 per cent to 260 per cent, thus reducing the danger of congestion and excessive breakage. When the universal or intermediate drive type of conveyor is used, the second conveyor undergoes a similar increase in capacity by changing either the diameter or the pitch of the flights or rotating the shaft at a slightly greater speed.

A point very often considered of vital importance is the relation between conveyor dimensions and breakage of the coal. Excessive degradation, because of its adverse effect upon combustion, is undesirable and should be eliminated wherever it is encountered. In order to insure the free and even flow of hard coal through a screw conveyor it is essential that all surfaces be smooth, free from all unnecessary projections, and that there be no constriction of cross-section or obstacle which might cause congestion. The experience of a number of designers indicates that screw flights should possess square edges in order to obviate any wedging of particles between the worm and tube walls.

In cases of bin conveyors having screws with identical dimensions both in the conveyor tube and extending into the bin, the danger of overfeeding is sometimes acute due to the fact that the capacity of the extension worm is greater than that of the conveyor itself. Under such conditions congestion at the inlet, excessive degradation, and possibly complete mechanical stoppage of the conveyor may result. The tendency to overfeed is influenced by the length of exposed screw and the relation between flight dimensions and coal size. The experiment given illustrates this point.

A bin feed conveyor having a 2½ in. inside diameter steel tube and fabricated screw with 1⅞ in. diameter and pitch was installed at an angle of 22 deg to the horizontal. The worm was allowed to project various distances, and after each change 500 lb of coal were conveyed. The accompanying table gives the results obtained:

PROJECTION (IN.)	SIZE COAL	DEGRADATION, PER CENT	OPERATION
15	Buckwheat	6.0	Overfeeding
10	Buckwheat	2.4	Slight Overfeeding
5	Buckwheat	1.0	No Overfeeding
15	Rice	2.0	No Overfeeding
10	Rice	1.2	No Overfeeding
5	Rice	0.8	No Overfeeding

This experimental work is obviously applicable only to the conveyor described, but from it can be drawn the general conclusion that where no special

pick-up worm is used, the length of exposed flights must be maintained at a minimum.

Although the clearance between conveyor screw and tube is often considered as a major factor in degradation, an actual survey has shown that this is not necessarily true. For example, clearances in the conveyors of stokers approved by Anthracite Industries, Inc. vary from 7 per cent to 50 per cent of the screw diameter. In conveyors with clearances ranging from 3/64 in. to 5/8 in. degradation of buckwheat size anthracite has been found to be reasonably limited. On the other hand, the fact that clearance is a contributing factor is shown by the following experiments in which all other factors were held constant:

Experiment No. 1: Bronze coal tube 2 9/16 in. inside diameter stainless steel fabricated worm 2 1/16 in. outside diameter. By using a smaller tube clearance was reduced

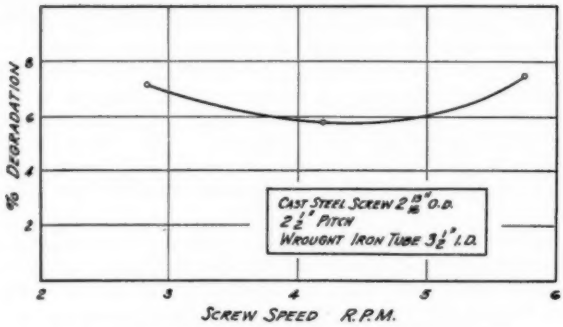

FIG. 3. EFFECT OF WORM SPEED ON DEGRADATION

from 0.25 in. to 0.047 in. This resulted in a 20 per cent reduction in the amount of undersize formed.

Experiment No. 2: In this case a cast-iron tube 3½ in. inside diameter was used in conjunction with two cast steel worms of approximately the same proportion but different diameters. By increasing clearance from 0.125 in. to 0.344 in., 34.6 per cent of the breakage was eliminated.

In these experiments buckwheat size anthracite screened between 0.312 in. and 0.562 in. diameter round mesh was used. The coal had identical characteristics in each case.

Although lacking in completeness, the foregoing evidence indicates a tendency toward maximum degradation at a point between zero clearance and clearance equal to the size of particles being moved. This maximum point is probably that at which there is the greatest tendency for wedging of coal between the conveyor flight and tube walls. Further study of this problem will be well worth the effort to the stoker industry.

The effect of conveyor speed is indicated in Fig. 3. A 3½ in. inside diameter wrought iron tube with a 2 13/16 in. outside diameter cast steel screw having a pitch of 2½ in. was allowed to feed 23 lb per hour at various speeds

of rotation. This was accomplished by the use of a bin conveyor which was operated at constant feed. Over the range explored variation of the speed of rotation even up to 100 per cent resulted in no appreciable change in degradation. The rate of rotation of anthracite screw conveyors at rating varies from 1.2 to 9.5 rpm. Table 1 shows approximately the limits ordinarily encountered in the dimensions of small anthracite screw conveyors.

CONVEYOR MATERIALS

There is a marked tendency toward the use of corrosion resistant steel shafts and flights in anthracite conveyors, particularly at the retort end. It has been definitely shown that chemical action rather than mechanical abrasion is responsible for the major part of conveyor deterioration. Experience indicates

TABLE 1—APPROXIMATE LIMITS IN DIMENSIONS OF SMALL ANTHRACITE SCREW CONVEYORS

	TUBE I.D. IN.	SCREW O.D. IN.	RATIO DIAMETER/PITCH	CLEARANCE, IN.	RPM	FLIGHT THICKNESS, IN.	SHAFT DIAMETER, IN.
Maximum.....	4	3 7/8	1.23	5/8	9.5	3/8	1 1/8
Minimum.....	2	1	0.5	1/16	1.2	3/32	1/2

that it is the draft and temperature conditions in the conveyor, rather than the number of tons of coal conveyed, which determine its life.

Moisture adhering to the surface of anthracite often contains small amounts of dissolved sulphur compounds which may under certain conditions cause corrosion of metals. When coal passes through the conveyor toward the retort it may encounter the following conditions: (*a*) sufficiently high temperatures to vaporize both the water and acids adhering to the surface of the coal; (*b*) draft conditions which allow the concentration of these vapors to build up by preventing (or retarding) their escape into the retort; (*c*) in some cases draft conditions may be such as to actually force corrosive vapors back from the retort.

Corrosion is encountered when conveyors are allowed to remain filled with coal during long periods of shut-down such as during the summer months. Worms operating all year round very often outlast those which are not used during the summer.

From the foregoing it is obvious that the remedy for most of the stoker conveyor corrosion lies in: (*a*) the use of acid resisting materials; and (*b*) modification of design to cause the dilution and rapid expulsion of vapors from the conveyor tube. A combination of the two is very often used, since it is widely recognized that, aside from corrosion, *gassing back* at the hopper or bin is a source of annoyance to the consumer.

The simplest and most universally used method for the sweeping of vapor from the conveyor consists of a connection or *cross-over* between the air duct

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and conveyor tube in such a manner as to impose the necessary amount of air pressure upon the latter to cause a continuous flow of air into the retort. Provision must be made at the same time to limit the amount of air which flows back toward the bin to a minimum. This is accomplished by intentionally causing a temporary congestion of coal so as to form a plug at a point between the conveyor inlet and the air cross-over.

A survey of the manufacturers of approved anthracite stokers gives an interesting insight into the various materials used in coal conveyors. For example, approximately 45 per cent of the manufacturers use fabricated steel worms, while 30 per cent use fabricated corrosion resistant steel (usually approximately 20 per cent chromium), and the remaining 25 per cent, cast steel. The distribution of materials used in coal tubes on the other hand is approximately

Wrought Iron	45 per cent (includes steel pipe)
Cast Iron	30 per cent
Bronze	25 per cent

In many cases several of these materials are used in the same stoker. For example, in a bin feed conveyor the major portion of the tube outside the furnace may be wrought iron pipe, with that part extending into the furnace of bronze. Likewise, a fabricated worm may be constructed of ordinary steel with that portion within the furnace being of chromium steel. The advantage of this latter procedure is readily recognized when it is realized that the formation of vapor in coal conveyors may be greatly reduced, but can rarely be completely eliminated.

Many other materials have been used experimentally for coal screws. Some, such as monel metal and bronze, have been found to resist corrosion but to be subject to abrasion. Others (stellite, tantalum, and vanadium steel) are at present too expensive. Difficulty due to warpage of cast iron has prevented extensive work with this metal. Manganese steel and Swedish steel were found to be subject to excessive wear. Plated and otherwise metal-surfaced screws have thus far been inferior to ordinary steel. Finally, several experimenters have tried vitreous enamel with success as far as corrosion and abrasion are concerned, but mechanical difficulties resulted in cracking and chipping.

It is of interest to note that the majority of anthracite conveyor screws are constructed of fabricated steels. The advantages claimed are: *a*. Greater accuracy obtainable. *b*. Square outer edges. *c*. Less wear at edges. *d*. No fillets necessary. *e*. Greater ease in splicing.

The development of improved methods of production is also exerting an influence in favor of fabricated screws. Greater structural strength is now possible by processes which wrap the flight onto the shaft in such a manner as to grip the latter tightly and uniformly at all points.

A conveyor not previously mentioned because its use is somewhat restricted is the plunger type. It is, however, being used very successfully in one well-known anthracite burner both for hopper and bin feed. The plunger, when

viewed from above, resembles a series of contiguous truncated cones having a common axis, while from the side it has a saw-tooth profile.

ANTHRACITE BURNER RETORTS

The combustion performance of an anthracite stoker is largely dependent upon the design and construction of its retort. The shape of the latter is highly important because of its relation to air distribution and the shape of the burning zone. The method of transition from conveyor to retort is a significant consideration from the point of view of coal distribution. The

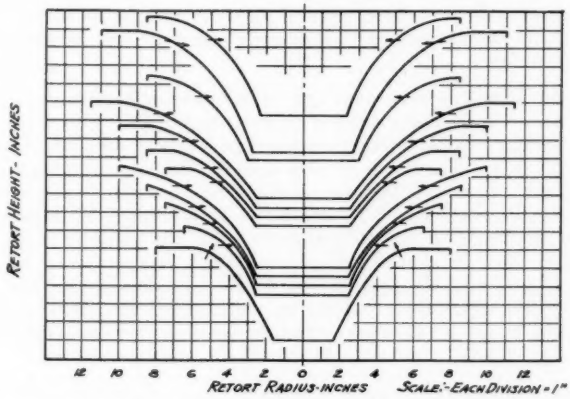

FIG. 4. TYPICAL RETORT CONTOURS

number, shape and distribution of tuyeres play an important part in the structure and consistency of the fuel bed. Each of these factors is essential, but it is only within limits that one of them can be varied to make up for deficiencies in another.

In order to maintain the uniformity and stability characteristic of a properly operating anthracite stoker fire, it is necessary that coal be distributed at an equal rate to all parts of the retort circumference. As shown in Fig. 1 the coal must be diverted from its horizontal travel through the conveyor to a vertical flow into the retort. Since this 90 deg turn must be made against the weight of a column of coal already in the retort (often over a foot in height) it is highly important that everything be done to minimize compression. It is at this point that 50 to 75 per cent or more of the degradation takes place. For this reason adaptor castings between the coal worm and the bottom of the retort are so designed as to gradually increase 100 to 200 per cent in cross-sectional area. These adaptors are usually referred to as the *gooseneck*

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or *throat* of the retort and should be so designed as to eliminate projections, large or small, which might interfere with the flow of coal.

In addition to the proper design of adaptor castings, the point at which the conveyor screw ends has a marked effect upon the distribution of coal in the retort. Unfortunately, although quite critical on most stokers, no generalization can be made.

In the combustion of anthracite, ignition takes place without preliminary coking. The small amount of volatile matter present is probably not released in appreciable quantities until burning actually starts. For this reason retorts of a decidedly shallow nature have usually been found to give best results. In order to bring about combustion in the retort at the rapid rate required, sufficient air for ignition is brought in contact with the fuel as soon as possible. In like manner ash must be eliminated as soon after it is burned out as is consistent with the maintenance of a satisfactory air seal around the upper edge of the retort.

For reasons already enumerated, one of the outstanding features of anthracite retort design is the ratio between diameter and depth. Thirty-two different models of anthracite stoker retorts included in a recent survey had an average ratio of 3.67 to 1. The depth in this case is taken as the distance between the top of the retort and the top of the throat casting. The ratio in the majority of cases was between 2.7 and 5.2 to 1, indicating the latitude possible. An idea of the shape of typical anthracite retorts is given in Figs. 4 and 5. Fig. 4 shows the actual vertical cross-sectional contour lines for different anthracite stokers varying from 13 in. to 23 in. in diameter. Fig. 5 shows the limits within which these various contours occur when they are corrected for variations in throat diameter. The shaded area is that within which the retorts of Fig. 4 fall.

The combustion performance of anthracite retorts is measured in terms of the number of pounds of coal efficiently burned per hour per square foot of horizontal projected grate area, the latter being arbitrarily considered as that falling in a plane bounded by the circumference of the retort. The actual *active* area is somewhat smaller and is ordinarily determined by the diameter of the circle formed by the outer tuyere or tuyeres. For the majority of retorts the active area falls between 35 and 50 per cent of the grate area and averages approximately 42 per cent. Thus, 58 per cent of the average (projected) retort area is devoted to the dead plate. This inactive ring over which the nearly consumed fuel passes serves a double purpose. Primarily, it keeps the remaining combustible in contact with the air stream in order to maintain combustion until the rate of heat loss lowers the temperature below the ignition point. This is necessary since, unlike bituminous coal, anthracite will not continue to burn after it is removed from the air stream. The second function of the inactive ring is to maintain a seal of ashes around the fire to prevent the escape of excess air.

The air ports or tuyeres in an anthracite retort should be so located as to permit a finely divided and low velocity air stream to penetrate the fuel bed as thoroughly as possible. The passage of pieces of coal through alternate zones of air blasts and no air causes the rapid formation of clinker. Likewise

anything which tends to cause *channeling* of the air either through the fuel bed or along the edges of the retort will result in the same difficulty. For this reason it is important that the windbox be so constructed as to deliver air at an equal pressure to all points at the same horizontal level in the retort. The velocity and amount of air as it enters the fuel bed on the other hand may decrease proportionately as the fuel bed becomes thinner. Fig. 6 shows cross-sections of various types of tuyeres used in anthracite retorts. These tuyeres may be divided into two general classifications: (*a*) the plate pinhole type which consists of relatively thin castings through which tuyere holes are drilled or cored; this classification includes both the segmental design in which a number of identical castings (usually 6 or 8) are arranged together to form the retort surface, and the type in which the entire active retort surface is in one piece; and (*b*) the ring type in which rings of increasing diameter are

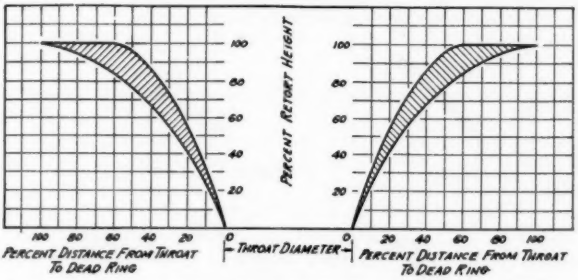

FIG. 5. RETORT CONTOUR LIMITS WITHOUT REGARD TO THROAT DIAMETER

placed one above the other so as to form the grate surface with annular tuyeres between each ring. Another tuyere used by one manufacturer is designated in Fig. 6 as the deflector type. Here air passes between a series of blank segmental plates and impinges upon deflector castings which spread it parallel to the plates. Sufficient turbulence is thus caused to make the air diffuse evenly into the fuel bed.

Tuyeres may also be classified in accordance with whether or not they permit the passage of fines into the windbox. For example, the pinhole type tuyere usually permits a certain amount of fine coal to drop through in spite of the opposing air pressure. On the other hand, the step type ring tuyere is effective in preventing this occurrence. Whether or not fines drop into the windbox is relatively unimportant since in the first place the more fines removed the less fly ash there will be, and secondly it is usually fairly simple to devise a suitable automatic or manual fines release.

A study of 12 approved anthracite stokers shows that the actual tuyere openings average very close to 0.5 sq in. per pound of coal at rated capacity. It is improbable, however, that this should be considered a basic design principle, since better performance is nearly always possible at lower combustion rates. It represents an average lower limit for good performance. The use

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of a greater tuyere area, if practical means could be devised to do so, would probably improve combustion conditions, particularly if it were accomplished by increasing the number of air ports.

RECENT DEVELOPMENTS IN ANTHRACITE STOKERS

A number of very interesting developments relating to stokers have been recently worked out by Anthracite Industries Laboratory and carried to various points beyond the experimental stage. Four of these are particularly outstanding, and as they are indicative of what may be expected in the near future, it will probably be of interest to describe them briefly.

The automatic ignition of anthracite fires has for many years been considered impractical. Nevertheless, it has recently been put upon a commercial basis

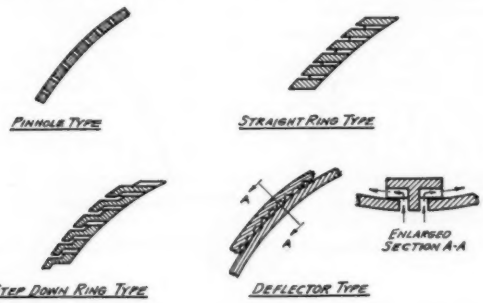

FIG. 6. TYPES OF TUYERES USED IN ANTHRACITE RETORTS

as a result of cooperation between the laboratory, a large manufacturer of electrical equipment, and a control manufacturer. Stoker manufacturers may now purchase a complete resistance unit for insertion in place of the lower part of the inside retort surface. This resistance unit carries the manufacturer's standard production guarantee. When used in conjunction with the proper control it permits the lighting of an anthracite burning stoker by means of a push button located at any convenient place in the house. Obviously, it may also be wired to an automatic control which causes the fire to relight whenever required. The current consumption is about ½ kwhr per operation.

It will be noted that as contrasted with bituminous stokers, the small anthracite burners referred to in this paper are equipped with circular retorts. The problem of coal distribution has been the main stumbling block which has stood in the way of progress upon anthracite retorts of other shapes. After an intensive study of the coal distribution problem, a rectangular retort was developed which spilled ashes off its four edges. It was found, however, that it is possible to impart sufficient sideward thrust in a narrow, rectangular retort

to enable the construction of a side feed burner having a length at least double its width. This has opened up a completely new field to domestic anthracite stokers, for as long as retorts are limited to a circular shape, it is impossible in many cases to provide sufficient input in long, narrow boilers. An additional advantage of the new rectangular retort is the fact that it may be made of cast-iron laminations in such a manner that it may be varied in length at the time of installation in accordance with field conditions encountered. Unfortunately, at the time of writing this paper it is not possible to disclose construction details. However, it is hoped to make them available to all stoker manufacturers in the near future.

The development of a practical and successful vertical ash conveyor has opened new possibilities for the solution of the ash removal problem. This conveyor is applicable to existing stoker ash removal systems. It carries the ashes vertically to whatever height necessary, while an overhead horizontal conveyor removes the material to any desired point. This enables removal of ashes from the cellar or to ash containers, pits, or bins located in the cellar or outdoors.

A new development of considerable interest and now on the market is a small stoker designed for the combustion of barley size anthracite. This stoker has a retort 7 in. in diameter and is extremely simple and inexpensive in construction. It has a combustion rate which may be varied from about ¼ lb per hour to nearly 3 lb per hour. It is used principally for automatic domestic water heating, but is being adapted for use in kitchen ranges, space heaters, and other applications. Combustion rates up to 5 and 7 lb per hour are obtainable by the use of slightly larger retorts. It is quite probable that the same design principles used in this stoker will be applied to larger sized units for house heating. When the potential market for stokers is realized, it does not take very much imagination to foresee the tremendous possibilities of a low priced unit such as this made largely of stampings and die castings.

CONCLUSION

The domestic anthracite stoker industry has completely emerged from an early period of trial and error engineering, during which time many important lessons have been learned. It is with the hope of diffusing some of the knowledge gained by various experimenters that those design features of particular importance in conventional anthracite burners of small size have been discussed at some length. That the future holds much promise for this industry from the angle of consumption is shown by the steadily increasing demand in spite of strong competition and other adverse circumstances.

That much may be expected along lines of technological advance is confirmed by the many new devices and inventions already waiting to be applied by aggressive manufacturers.

ACKNOWLEDGMENT

Much information used in this paper was supplied by the following individuals to whom the author wishes to express his sincere appreciation: G. Crocker, C. Crowley, C. H. Hall, G. M. Peltz, A. Rogers, and M. A. Young.

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Part II. Small Stokers for Bituminous Coal

COAL has long been the fuel predominantly used for heating in the United States. Bituminous coal and anthracite heat over 85 per cent of the one- and two-family residences of the country. Between 15 and 20 per cent of the total production of bituminous coal, or 60 to 80 million tons of a normal production of 400 million tons, is used for domestic heating.

Most of the bituminous coal used for heating has been of the larger, so-called domestic sizes, fired by hand in central heating plants and stoves. Fired in this way, coal has the advantages of simplicity and dependability of equipment. Given the coal, which is produced in 28 states and widely distributed to the remainder, and the simplest type of fireplace, stove, furnace, or boiler and with no dependence on the availability of electric power or other utility, a home or building can be heated with little possibility of failure.

The simplicity and dependability of bituminous coal as a fuel for comfort heating have been attended by certain disadvantages. Among these are (1) dirt and smoke, (2) the tendency toward non-uniformity of heating, and (3) inefficiency in the utilization of the potential heat available in the coal. All of these difficulties may be greatly reduced, if not entirely eliminated, by intelligent methods of firing but this generally introduces a fourth difficulty, that of increased frequency of attention. Despite these disadvantages of bituminous coal and despite the availability of automatic firing devices for oil and gas earlier than for coal, coal held its position because it is, in most parts of the country, an economical source of heat.

A need for an automatic device to fire bituminous coal in heating equipment without the disadvantages of hand-firing has existed for many years. That this need was not met by manufacturers of stokers that were used under large heating and power boilers is explained by the fact that an entirely new type of engineering and merchandising was required. Whereas the firing equipment for larger boilers is more or less individually designed and adapted to each installation, the possible sales price of the smaller equipment could not absorb such individual engineering expense. New methods of mass production and mass merchandising of equipment that could be sold and installed with a minimum of time to each job were demanded.

This demand has been met and an entirely new group of manufacturers of small stokers has arisen. Although there is some overlapping in the fields of the largest sizes of one group with the smallest sizes of the other, the fields are quite distinct. The early installations of the small stokers were in the larger types of heating boilers in commercial buildings and larger homes, but the range of sizes extended downward until today the largest number of sales of stokers are those having a maximum coal burning rate of approximately 20 lb per hour. Difficulties with the early installations of the smaller sizes have been reduced by refinements produced by research and engineering developments and the satisfaction that the stokers give is proved by their increasing sales over the past few years.

From 9571 in 1932, the number of stokers sold increased to 101,808 in 1937. Of the total in 1937, 87,451 or 86 per cent were of the residential size, burn-

ing 60 lb of coal or less per hour; stokers burning bituminous coal numbered 78,377 and those burning anthracite 9074 in this class. This demand for stokers has been a part of the increasing public demand for automatic heat as the sale of oil burners and gas burners increased markedly in the same period. The sale of stokers in July, 1938, was 9061 units as compared with 10,689 oil burners.

STOKERS OPERATE ON UNDERFEED PRINCIPLE

Although a considerable number of the small stokers designed and used for burning the non-agglomerating coals of the far West are of the overfeed type as described by Yancey,[5] and some overfeed stokers have been made and sold in the remainder of the country, most of the stokers of the residential

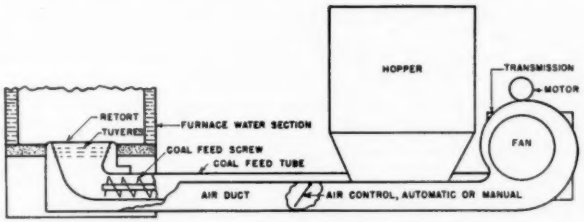

FIG. 7. SCHEMATIC DIAGRAM OF CONVENTIONAL DOMESTIC UNDERFEED STOKER

class and larger are of the underfeed principle. Fig. 7 shows diagrammatically the principal parts and general method of assembly of the conventional small stoker. The fractional horsepower motor has its speed reduced to 1 to 2 rpm at which the feed screw revolves by a transmission that may be of the gear or hydraulic type. The screw moves the coal from the hopper, which has a capacity of 250 to 400 lb of coal, through a cylindrical feed tube into the bottom of the retort shown at the left of the figure.

The retort may be either round or rectangular. Round retorts are more common in the smaller sizes of stokers and rectangular retorts are the rule for large stokers. The coal moves upward in the retort past the ports or tuyeres distributed in one to four rows around the inside of the retort near the top. A row of tuyeres is also commonly provided around the outside of the retort at the top.

Air for combustion is provided by a centrifugal fan, the rotor of which may be mounted directly on the shaft of the motor or driven through a V-belt. A damper, which may be manual or automatic, is provided in the air duct or on the inlet of the fan for adjustment of the rate of supply of air.

Because pieces of foreign material that will catch between the flight of the feed screw and the entrance to the feed tube may be placed in the hopper with

[5] Burning of Various Coals Continuously and Intermittently on a Domestic Overfeed Stoker, by H. F. Yancey. (Report of Investigation 3379, *Bureau of Mines, January*, 1938.)

the coal, a removable plate is generally provided at the junction of the hopper and the feed tube for removal of the obstructing piece. A shear pin, an electrical-overload switch, or a hydraulically-operated switch in hydraulic transmissions or combinations of these is provided for protection of the motor from overloads caused by jamming of the feed screw. To minimize or prevent arching of wet coal in the hopper, many stokers are provided with agitating devices in the hopper.

For further discussion of the details of the design of small stokers and the principles underlying the trends in design, the reader is referred to a recent paper by Guthrie.[6]

INSTALLATION OF STOKERS

Stokers are normally installed with the top of the retort at the level at which the grate was or would be. A steel plate may be used between the retort and the walls of the combustion chamber or more generally the space below the level of the hearth is filled with bricks and sand. Above the steel plate or filling, the hearth is formed with plastic refractory. In larger installations, cast-iron dead plates are used instead of refractory and in some of the largest installations dumping side grates are used.

Because one of the principal advantages of a stoker installation over hand firing is the possibility of combustion without smoke, the installation should be made with ample distance between the hearth and the heating surfaces and with ample combustion volume to obtain proper mixing and prevent smoke. Standards for setting heights and combustion space requirements that are acceptable to stoker manufacturers, boiler manufacturers and smoke regulating departments have not yet been provided. The *Steel Heating Boiler Institute* sets a maximum limit of 60,000 Btu per cubic foot of combustion space per hour and stoker manufacturers state that this is acceptable. They report installations where this rate of heat liberation has been exceeded with satisfactory operation. On the other hand, limits as low as 35,000 and 40,000 are set by some smoke regulating departments. Arrangements are now under way for an experimental investigation to determine the proper limits for various types of coal.

SIZING OF STOKERS

With stokers, as with all types of firing equipment, care must be exercised to have the equipment properly sized to fulfill the heating requirements. *The Stoker Manufacturers Association* has recently adopted a Uniform Stoker Rating method. The standard formula is:

$$\text{Stoker size in pounds of coal (burning rate per hour)} = \frac{\text{Load in Btu/hour}}{\text{Heating value of coal as burned} \times \text{over-all efficiency of stoker and boiler or furnace}}$$

[6] Small Underfeed Single-Retort Stokers, by B. M. Guthrie. (*Transactions ASME*, Vol. 60, 1938, p. 5-10.)

For warm-air furnaces the load is stated as the Btu heat loss of the space to be heated and for steam and hot water systems the load is calculated as the product of the installed radiation or its equivalent by the usually accepted heat emission factors of 240 and 150 Btu per square foot per hour, respectively, and a piping and pick-up factor of 1.33. The pick-up factor is subject to change to higher values for unusual conditions.

The value for over-all efficiency of the installation suggested is 65 per cent for burning rates up to 100 lb of coal per hour and 70 per cent for rates above this. These values are only suggested average values and should be decreased or increased dependent on the particular conditions of the installation.

PROCESS OF COMBUSTION

Through research conducted by Bituminous Coal Research, Inc., at Battelle Memorial Institute, the process of combustion of bituminous coal on the small underfeed stoker has been subjected to a detailed experimental analysis. In a series of reports by Sherman and Kaiser,[7] by Sherman,[8] and by Barnes,[9, 10] the results and conclusions have been reported in detail.

Briefly stated, these conclusions are: Combustion does not proceed by the strictly underfeed principle but by a combination of underfeed, crossfeed and overfeed principles. With most bituminous coals, the caking and coking are so favored by the rapid rates of heating in the fuel bed that masses of coke much larger than the individual pieces of coal fed to the stoker are formed. The air from the tuyeres cannot penetrate to the center of the mass of coke and combustion occurs in a ring around the periphery of the retort rather than over the entire area of the retort. The coke rises in the center, breaks up and falls into the combustion zone. The ash as released is pushed onto the hearth around the retort where it is subjected to the high temperature from the combustion zone and is fused into a ring of clinker for removal with tongs.

SELECTION OF COAL FOR UNDERFEED STOKERS

Performance tests of various typical coals have been included in the program of research of Bituminous Coal Research, Inc., and have been reported by Sherman, Kaiser, and Limbacher,[11] Limbacher and Sherman,[12] and Sherman.[13] The results of these tests have shown that of the various characteristics of bituminous coals, such as volatile content, ash content, ash clinkering tendencies, caking and coking characteristics, and size, the coking characteristics and the clinkering tendencies were the most important in determining the satisfactory performance of a coal.

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[7] Combustion of Bituminous Coal in Small Underfeed Stokers, by R. A. Sherman and E. R. Kaiser. (*Transactions Coal Division, AIME* Vol. 130, 1938, p. 388-407; Information Bulletin No. 2, *Bituminous Coal Research, Inc.*, 1937.)

[8] Automatic Residential Heating with Bituminous Coal, by R. A. Sherman. (*Proceedings*, 22nd Fuel Engineering Conference, Appalachian Coals, Inc., 1938.)

[9] Fundamentals of Combustion in Small Underfeed Stokers, by C. A. Barnes. (Technical Report No. 4, *Bituminous Coal Research, Inc.*, 1938.)

[10] Some Aspects of Combustion in Small Underfeed Stokers, by C. A. Barnes. (Presented at joint meeting of Fuels Division, *ASME* and Coal Division, *AIME*, October 1938.)

[11] The Relation of the Size of Bituminous Coals to Their Performance on Small Underfeed Stokers, by R. A. Sherman, E. R. Kaiser and H. R. Limbacher. (Technical Report No. 1, Part II, *Bituminous Coal Research, Inc.*, 1937.)

[12] The Performance of Several Types of Bituminous Coals on Small Underfeed Stokers, by H. R. Limbacher and R. A. Sherman. (*Bituminous Coal Research, Inc.*, 1938.)

[13] Preparation of Stoker Coal, by R. A. Sherman. (*Coal Heat*, Vol. 34, August 1938, p. 5-8.)

The coking characteristics are of utmost importance because, although practically all coals form more or less coherent masses of coke that make a ragged-appearing, non-uniform fuel bed, if the coking is too strong certain difficulties may result. The principal difficulty is a low rate of burning and slow response to demands for heat with, in extreme cases, loss of ignition and extinction of the fire. The maintenance of a low excess of air has been found favorable toward the decrease of difficulties with coke formations, not because this reduces them but because it favors the maintenance of a strong burning zone and maintains ignition.

The usual method of removal of ash from small stoker-fired installations is as a clinker. The ash must fuse, therefore, at the temperatures reached in the fuel bed of the stoker but must not become so fluid as to run down into the retort. Barnes [14, 15] has shown that temperatures of 2400 to 2800 F are reached in the fuel bed of typical small stokers in continuous operation. With intermittent operation, as in heating installations, the maximum temperature reached over prolonged periods may be several hundred degrees lower than the maximum Barnes found.

Experience has shown that coals whose ash-softening temperatures are in the range of 2200 to 2500 F are well suited to use on the clinkering type of stoker. Cleghorn and Helfinstine [16] and Cleghorn [17] have shown that Iowa coals whose ash contents are 15 per cent and higher and whose ash softening temperatures are 2000 F and lower can be used on the small stoker and many successful installations using this type of coal confirm this finding. Coals with ash-softening temperatures as high as 2700 F have also been successfully used but difficulty is to be expected with such coals from failure to make clinker in periods of prolonged mild weather.

The size of the coal furnished for small stokers can be controlled by the producer and the importance of this characteristic has been the subject of investigation by Bituminous Coal Research, Inc., by coal producers, and by stoker manufacturers. The conclusion reached as to the top size of coal is that for the residential stoker, burning less than 60 lb of coal per hour, it should not be greater than 1¼ in.; many producers have established the top size at a lower value, as 1 in., ¾ in., and as low as 5/16 in.

The lower size limit has been found by Bituminous Coal Research, Inc., to be of particular importance in the combustion process only with the strongly coking coals. With these coals it is often an advantage to remove the coal smaller than 10 mesh, ¼ in. or ⅜ in. as the coking tendencies will be reduced either because of the removal of a strongly coking fraction of the coal or because of increased porosity of the bed. The effect of the removal of the finer sizes would undoubtedly be greater if it were not for the crushing action that occurs in the feeding of the coal from the hopper to the retort.

Close sizing of coal has other advantages apart from whether or not it improves the combustion performance. It reduces the segregation of the sizes that occurs in the handling of the coal, it improves the appearance of the coal, and it makes the coal more amenable to dust-proofing with oil or other materials.

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[14] Loc. Cit. See Note 9.
[15] Loc. Cit. See Note 10.
[16] Use of Iowa Coal in Domestic Stokers, by M. P. Cleghorn and R. J. Helfinstine. (Bulletin 134, *Iowa Engineering Experiment Station*, 1937.)
[17] Iowa Coal as a Domestic Stoker Fuel, by M. P. Cleghorn. (Presented at Joint Meeting of Fuels Division, *ASME* and Coal Division, *AIME*, October 1938.)

A disadvantage of close sizing is that it necessitates the production of a small size of coal that is difficult to handle and dispose of profitably. This results in the necessity of a higher price for the sized stoker coal than would be possible for the coal without the smaller sizes removed. The recent trend by producers has been toward the production of the closely-sized stoker coal although some adhere to the production of stoker coal without the fines removed where experiment has shown that no advantage in performance results from their removal.

The desire for cleanliness is one of the impelling motives for the purchase of a stoker for heating purposes. The coal that is used must, therefore, be free from dust. Producers recognize this and practically all of the coal marketed for small stokers is treated to eliminate dust in handling. A program of research has been sponsored at Battelle Memorial Institute on the dustless treatment of coal with oil which is one of the materials widely used for this purpose. Papers by Sherman and Pilcher [18] and by Pilcher [19] present some of the results of this investigation. They show that the type and amount of oil required for permanently effective treatment varies with the rank and the size of the coal but that all coals can be effectively treated with some of the economically available petroleum products.

SPECIAL FEATURES OF STOKERS

In addition to the essential features of small stokers that were described briefly at the beginning of this paper, a number of special features are used by one or several manufacturers. The description and comment on these features have been placed after the discussion of the process of combustion and of the relation of the characteristics of coals to their performance that the purpose and action of these features might be more clearly understood.

AUTOMATIC AIR CONTROLS

The centrifugal fan that is used on most small stokers has the well-known characteristic that for any fixed setting of a controlling damper the volume or weight of air delivered varies widely as the resistance against which the fan works varies. The variations in resistance of the fuel bed that occur in the intermittent operation of a stoker used for heating will, therefore, result in wide changes in the rate of air flow although the rate of feeding coal may remain constant. Many manufacturers have equipped their stokers with dampers that function automatically to vary the natural characteristics of the fan. These dampers may function by static pressure or by velocity pressure of the air in the duct to maintain a constant flow of air regardless of the resistance of the fuel bed or to increase the flow of air as the resistance of the bed increases and to decrease the flow as the resistance decreases.

Fig. 8 shows the type of control obtained with one air control as reported by Limbacher [20] as a result of research for Bituminous Coal Research, Inc. This was of the type that was designed to maintain a constant flow of air and

[18] An Experimental Investigation of the Use of Oil for the Treatment of Coal, by R. A. Sherman and J. M. Pilcher. (*Transactions, ASME*, Vol. 60, 1938, p. 97-111.)

[19] Oil Treatment of Coal, by J. M. Pilcher. (Paper presented at Conference on Coal Utilization, University of Michigan, 1938.)

[20] Performance of Automatic Air Controls for Small Underfeed Stokers, by H. R. Limbacher. (*Heating and Ventilating*, Vol. 35, 1938, p. 40-42.)

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the flatness of the curves shows that it functioned as designed up to the cut-off pressure of the fan. Limbacher's report showed that four of the five controls tested filled their purpose adequately.

FUEL BED DEPTH CONTROLS

Another method of attaining a similar end has been adopted by several manufacturers. A manually set damper is used for the control of the air but the flow is held substantially constant by control of the rate of feeding coal to maintain a constant resistance in the fuel bed. One stoker does this by the

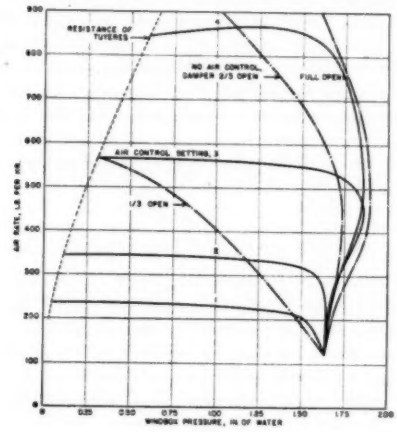

FIG. 8. CHARACTERISTICS OF AN AIR CONTROL INTENDED TO MAINTAIN A UNIFORM AIR DELIVERY AS THE WIND-BOX PRESSURE VARIES

use of separate motors for the fan and the coal feed; the coal feed motor is automatically stopped when the resistance of the bed rises to a predetermined and adjustable value and is started again when the resistance decreases to another value. A variation of this method is to use but one motor with a solenoid-operated clutch between the motor and the transmission to stop the coal feed. A discussion of the operation of this type of device is given in a recent report.[21]

FUEL-BED AGITATORS

As was discussed earlier in this paper, the coking characteristics of coal are quite important in the determination of the suitability of the coal for small stokers. To widen the range of choice of coals, manufacturers have designed agitating devices to break up the coke as it is formed in the bed and prevent the formation of large masses that might occasion difficulties. One of these is a mechanically operated poker that is inserted into the fuel

[21] Loc. Cit. See Note 12.

bed from the bottom of the retort at frequent intervals. Another type consists of a short screw extending diagonally upward in the retort from the back to the front. In a report,[22] results of a series of burning tests with one of these devices are given; the report presents the disadvantages and advantages of the device and concludes that the principle has merit for coals that are very strongly coking.

BIN-FED STOKERS

The screw-feed principle that is used on practically all of the small stokers not only permits placing the coal hopper at a distance from the firing door of the boiler or furnace that will allow easy access to the furnace but also permits the feeding of coal directly from the bin when desired. Some manufacturers build into the user's bin a hopper-type bottom and others use a rotating transfer screw in a flat-bottom bin to bring the coal to the entrance to the main feed screw. Several variations in the design of equipment are used and coal can even be made to make a right-angle turn in its flow from the bin to the heating unit. As with anthracite, the bin-feed principle has been well worked out and is successfully used in many installations.

AUTOMATIC ASH REMOVAL

As previously discussed, the ash remaining from the combustion of the coal deposits on the hearth of the furnace where it sinters or fuses into a clinker that can readily be removed with a pair of clinker tongs. Although this operation is readily done and is normally not necessary more than once each day, it is a feature of operation that prevents the conventional stoker, even with bin feeding, from being completely automatic. The development of a stoker that will remove the ash automatically and place it in a container in the basement or outside the house has received and is receiving the attention of a number of manufacturers. Because of the wide range of clinkering tendencies of the ash of bituminous coals, the attainment of a design that will be satisfactory for the entire range of coals is more difficult than for anthracite which has a much narrower range of ash-softening temperatures. One manufacturer, a subsidiary of a coal producing company, has, however, had on the market for a number of years a stoker that includes the principles of bin feeding and automatic ash removal. In the distribution of this stoker, it is restricted to the use of coal specified by the manufacturer; this method has simplified the problem of obtaining satisfactory performance.

CONTROLS FOR STOKERS

Controlling devices that will govern the operation of the stoker to maintain a fuel bed in condition to respond to a demand for heat, transmit the demand for heat to the stoker, and prevent overruns of temperature are essential accessories to a stoker-fired heating plant. The normal controls furnished are (1) a thermostat in the living quarters to start the stoker directly or through a relay as the room requires heat or to stop it when the requirements are satisfied, (2) a device to operate the stoker at such intervals

[22] Loc. Cit. See Note 12.

as required to maintain a fuel bed when heat is not required, and (3) a limiting device to prevent rates of heat liberation greater than can be transferred to the space to be heated.

Thermostats operating from room temperatures are the usual type for this service as used on hand-fired furnaces and oil and gas burners and require no comment. They include the heat anticipating and accelerating types.

The most frequently used device to maintain a fire is a time switch that operates the stoker for a few minutes at half-hourly or hourly intervals. This is not wholly satisfactory as coals vary in the length of operating period required to maintain a suitable fuel bed and even with one coal the length of the period varies from time to time. Combinations of time switches to turn the stoker on with a stack switch to turn it off when some set temperature has been reached have been used. These have not been completely satisfactory because the thermostatic elements do not retain their calibration when overheated or heated for long periods at high temperatures. Modifications that include a stack thermostat that turns off the stoker when a certain temperature rise rather than a certain temperature has been attained have been proved to have merit and a thermostat mounted in the door of the firebox to respond to furnace temperature has also been found suitable.

Limiting devices to prevent overruns of temperature that will demage the furnace or boiler or overheat the space are essential on stokers as on other types of firing equipment. The fact that a large mass of incandescent fuel may be left in the bed at the time the room thermostat is satisfied is conducive to overruns as this bed continues to radiate to the heat-absorbing surfaces of the combustion space. Stack or firing-door thermostats, aquastats, pressurestats, and bonnet thermostats all have been used as limiting devices for this service.

THERMAL EFFICIENCIES

The thermal efficiency of a stoker-boiler or stoker-furnace combination is of less importance to the home-owner than the degree of attention-free and trouble-free operation, but any calculations of cost of heating for any comparative purposes will depend on this factor which is of interest particularly to the heating and ventilating engineer. No general statements of the efficiency to be expected are of great value as the overall efficiency is as much a function of the boiler or furnace as it is of the stoker.

The values of 65 per cent for stokers burning less than 100 lb of coal per hour and 70 per cent for those larger that have been suggested by the *Stoker Manufacturers Association* have been cited. Yancey [23] reported overall efficiencies of a hot-water boiler fired with an overfeed stoker of 64 to 77 per cent for both continuous and intermittent operation when burning weakly coking coals suitable to the type of stoker. Cleghorn [24] obtained efficiencies of 50 to 67 per cent with an underfeed stoker and a steam boiler; variations were caused by the type and rate of operation and the type of coal. Sherman and Cross [25] give the results of field tests conducted on underfeed stokers in a hot-water and a steam boiler and a warm-air furnace with the conclusion

[23] Loc. Cit. See Note 5.
[24] Loc. Cit. See Note 17.
[25] Efficiencies and Costs of Various Fuels in Domestic Heating, by R. A. Sherman and R. C. Cross. (Technical Report No. 3, *Bituminous Coal Research, Inc.*, 1936.)

that a normally expected efficiency for the average conversion installation with average adjustment may be 55 per cent.

The factors that will govern the efficiency that will be obtained are the excess air maintained and the effectiveness of heat absorption in the heat-transfer surfaces of the heating equipment. If the heat-absorbing surfaces are not ample for the rates of heat liberation used, the stack temperatures will be high and the efficiency low even if a low excess air is maintained. On the other hand, adjustment of the air supply to excessive values will result in low efficiencies. The loss due to combustible in ash is practically nil with the clinkering type of stoker and losses due to unburned gaseous combustible in the flue gases should also be negligible. With properly coordinated heat-absorbing surfaces and rates of heat liberation and with an excess of air during operating periods of 50 to 60 per cent, corresponding to 10 to 11 per cent CO_2, efficiencies of 65 to 70 per cent should be obtained.

TABLE 2—DISTRIBUTION OF HEATING METHODS IN 1- AND 2-FAMILY URBAN HOMES OF THE UNITED STATES, AT CLOSE OF 1936

HEATING METHOD	NUMBER	PER CENT
Stove heated	6,547,658	45
Hand-fired coal furnace	5,926,263	40
Automatically-fired		
Oil	1,349,401	9
Gas	639,536	4
Stoker	219,403	2
Total	14,682,261	100

FUTURE STOKER MARKET

Of heating plants of every type above the residential size, it may be safely said that they may be included in the prospective stoker market where coal is available at prices competitive with other fuels. This is true because the possible savings due to the use of coal on a stoker over hand firing or over oil or gas in fuel alone or fuel plus labor will pay for the cost of the firing equipment.

Of residential heating plants, however, this is not true because the smaller the heating plant the more difficult it becomes to show a saving great enough to pay for the stoker. The cost of the stoker does not decrease at the same rate as the amount of fuel saved decreases. For example, assuming that the efficiency obtained with hand-fired bituminous coal is 45 per cent and that with a stoker is 60 per cent, a saving of 25 per cent of the fuel should be possible. Actually, on the small plant, the amount of coal used may not decrease because due to the greater ease of obtaining heat more heat will be used. If, however, the saving of 25 per cent is assumed, this will amount on a house using 8 tons of coal at $8 per ton to $16 per year. The stoker may be assumed to cost $250 installed; with fixed charges of 15 per cent, the annual charge is $37.50. The net cost to the owner for the convenience and comfort of the stoker is $21.50 per year.

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Although this premium for improved heating is less, in most localities, than for other fuels, it shows that the market for stokers is governed by the ability of the customer to buy. The percentage of the potential customers sold will depend on the merchandising ability of the industry.

No complete analysis of the potential market will be attempted in this paper because a survey for the entire country should be a summation of analyses for each city rather than from average figures for the entire United States. The data for individual cities were not available. Some conception of the possible market can, however, be gained from the data for the entire nation.

Data given in Table 2 show distribution of heating methods in 1- and 2-family urban homes of the United States.

On the basis of ability to buy, it is seen that of the present homes those heated by stoves can be dismissed as prospects. The entire remainder cannot be considered as prospects as this will depend on their income. Mean values of family incomes from the Brookings Institute 1929 survey and from the Federal Income Tax Returns of 1933 as accepted by the *American Gas Association* [26] in a survey on the economics of gas househeating showed that 25 per cent of the families had incomes greater than $2200 annually.

If it is assumed that families with incomes of $2200 or more can afford to pay the $21.50 annual premium for stoker-fired heating, then 25 per cent of the homes may be included in the potential stoker market. This amounts to 3,670,565 prospects. If this lower limit of income is correct, then all above can be considered as prospects on the basis of income for, as previously pointed out, the premium for stoker heating decreases as the size of the house and amount of fuel used increase. The potential market for competitive fuels is more limited than that for stokers because, in most districts, a premium for the oil or gas must be paid over hand-fired coal in addition to the fixed charges on the equipment.

From the total potential market must be deducted those already supplied with automatic heating leaving a balance of 1,462,225. This figure must also be adjusted by deduction of those homes that are in the far west and southwest where coal is not a competitive fuel and by an addition on the logical assumption that many of those installations now using oil and gas are prospects for stokers. No attempt has been made to make these adjustments.

The figures given include all one- and two-family homes whether owner-occupied or rented. Rented homes are prospects for stokers as well as owner-occupied homes although the sales effort required is undoubtedly greater.

This brief discussion of the market for stokers does not give any conclusive figures but it does show the method that should be used in each district to determine the sales possibilities. It indicates that the market for conversion installations in present homes is not limitless and may reach saturation within a relatively few years. Stokers at lower cost will extend the range of incomes that may be included in the market and more stokers that are fully automatic will increase their sale in that market that demands this type of operation.

The ultimate market for stokers has not been reached for new homes. Those new homes in which boilers and furnaces specifically designed for oil

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[26] A Study of the Economics of Gas Househeating, by A. M. Beebee. (Report of the House-heating and Air Conditioning Committee, *American Gas Association, April*, 1936.)

or gas are installed are quite definitely removed from the stoker market. However, a more conventional boiler or furnace fired with a stoker offers the owner the advantage that later conversion to other fuels is possible for the cost of the burner if circumstances should warrant the change.

SUMMARY

This paper has not attempted to cover fully the information that a heating and ventilating engineer should have on the subject of small stokers designed for burning bituminous coal. It has rather attempted to point out the highlights and include references to recent literature that will give those who require it the complete information required to insure satisfactory installations of this economical method of heating.

DISCUSSION

W. T. REID:[27] In the domestic underfeed anthracite stokers, is it not possible to burn anthracite on an overfeed principle, too? In the far West, where non-coking coals are available, the overfeed or crossfeed stoker has had considerable acceptance by the public, principally because of its very cheap cost.

Also, on the ash removal system with anthracite, in spite of the high ash fusion characteristics, does clinker ever form, and, if so, would it not interfere with the operation of the ash removal system?

P. A. MULCEY: To answer the first question, Yes, overfeed stokers have been used, particularly with the larger coal sizes. The most important difference between under and overfeed is in ash removal. Overfeed stokers are usually designed for use on hand-fired grates. Therefore, it is necessary to shake the grates and remove the ash in the usual way. Therefore, the convenience factor of the underfeed stoker is partly lost.

In answer to the clinkering question, yes, it is possible to get clinkering in an anthracite stoker, but it is extremely undesirable, and usually may be overcome by proper design. A stoker which causes clinkering can usually have its tuyeres redesigned in such a manner as to greatly reduce clinker formation. There may be variations in the type of clinkering. In many cases a mass of soft ashes which has just started to clinker appears on the dead ring. This may be quite conducive to good operation. In a properly functioning anthracite underfeed stoker no serious clinkering should occur. Normal ash-conveyor design will take care of the limited amount of clinker formed under average conditions as well as abnormal quantities.

COL. W. A. DANIELSON: Have we had enough experience yet with stokers to determine what their probable life will be; also during operation in periods of low heating, what difficulties have been experienced?

MR. MULCEY: This question should be answered from both standpoints. I will answer for anthracite, and Mr. Sherman may want to carry on for bituminous.

In anthracite stokers, we still have burners in operation which were installed some fifteen years ago, operating very satisfactorily. On the other hand, as I said in the paper, the matter of material selection has a lot to do with it. In some cases we run into corrosion difficulties with ordinary steel which may limit the life of

[27] U. S. Bureau of Mines, Pittsburgh, Pa.

certain parts of the stoker very seriously. On the other hand, by using stainless steel or other alloys we can extend the life considerably, ten times as long very often, in the case of coal screws, and it is here that stainless steel has been used most extensively. In some cases there has not been sufficient time to establish just what the life of such equipment is.

With respect to the retort, ordinary cast-iron in some cases will stand up for extremely long periods. In some applications, such as in bake ovens, the life of the ordinary cast-iron tuyere plate is short. However, alloys have been found which are satisfactory even in extreme applications. In general, I would say that we have not yet found the average life of the anthracite stoker.

During periods of low demand no particular difficulties are experienced with the anthracite stoker. The chief problem is that of keeping the fire ignited over long periods of no demand, which are often encountered in mild Spring and Fall weather. Though a stoker with buckwheat coal may bank successfully for several days, it is the usual practice to use a "hold-fire" control which causes the stoker to operate for short periods at various intervals.

R. A. SHERMAN: In regard to the question of life, experience has not been long enough to state definitely what the expected life will be. I think so far we must take a normal expectancy of, say, ten years. There will obviously be maintenance in that period.

I think you asked a question about the control in mild weather?

COLONEL DANIELSON: Yes.

MR. SHERMAN: That is, to a degree, a problem; there are difficulties with stokers for bituminous coal in mild weather, under some conditions. It is largely a problem of the type of control, and developments will come and many already have come. The normal type of control for holding a fire is one that turns the stoker on at intervals of 30 min to 1 hour and lets it run for a fixed time, which may be 3 min., 5 min., or more. It is obvious that there may be over-heating in periods of mild weather, and also it is true that this interval may not be long enough, and the fire may go out; this is, of course, a very bad thing.

We believe that a type of control in which the stoker is turned on at intervals of 30 min or 1 hour and is left on until some definite actions have occurred, such as a definite rise in flue-gas temperature or a definite rise in furnace temperature, and then is turned off, is much better than a fixed time, because the fire will be in different condition at different times. At one time it may require only 1 min, and at another time it may require ten. An average of 5 min may not be at all suitable.

W. C. KELLY: I think I could perhaps elucidate a little further on the question that was raised as to the life of the domestic stoker. There have been domestic stokers on the market for 15 years, many of which are still in operation. Since that start 15 years ago, there have been numerous improvements, as is indicated by the use of stainless steel. The stoker industry can only improve as rapidly as the products which go into its manufacture are improved by their respective manufacturers. The tremendous strides that have been made in the production of new types of steel and newer types of controls have made great improvement in the design and performance of the socalled small stoker, regardless whether it be bituminous or anthracite. Naturally over a long period of time there will be some maintenance. A conservative figure for maintenance and service probably would not exceed $1.00 per ton of coal on a small domestic stoker using an average good grade of coal. Bear in mind, however, that types of coal vary and maintenance cost varies with the type of fuel. On the larger types of stokers with a greater coal consumption maintenance will be reduced to approximately fifteen cents per ton. On the larger boilers and

stokers, maintenance probably would drop to from five to six cents per ton of coal and many are slightly under this point.

In regard to controls, these units have been greatly improved. When you go to a bin-feed type of stoker, where you have a bin that holds from 5 to 8 tons of coal, it is obviously necessary to shut off the stoker equipment should trouble develop or the fire go out. If this were not possible, the continued operation of the stoker equipment might entirely fill the furnace with unburned fuel. This need has led to the development of a control such as was mentioned by Mr. Sherman, which will at periodic intervals operate the stoker a sufficiently long period of time to create a rise in stack temperature. This rise, of course, indicates that the fire is maintained in proper condition and also this has a tendency, because of the added heat, to circulate the air in the heated area. At periodic intervals, the stoker will turn on. This interval is set in accordance with the type of fuel used and may vary from 30 min to 1½ hours. The control is so arranged that a refueling operation can not immediately follow a thermostatic operation; in other words, the interval is always counted from your last stoker operation. If this were not done, there would be a possibility of over-heating. Also, if your control indicates that a normal stack temperature is still maintained this fact makes it unnecessary for the refueling mechanism to operate and that interval is past and the control returns to zero setting. If, however, there has been a drop in the stack temperature a unit will refuel until a reasonable increase in stack temperature is indicated. This increase is usually set from 75 F to 100 F, depending on the type of furnace and the stack temperatures. Obviously with a two or three phase steel boiler your stack temperatures would be lower than they would be with a small round vertical cast-iron sectional boiler, where you get an immediate stack rise with reasonably high stack temperature.

These improvements in the products themselves and in the controls have only come as rapidly as the knowledge of the industry increased and this increase in acceptance and knowledge has increased, as is attested by the fact that there are approximately 400,000 of these units in operation.

L. P. HYNES: I have been using an anthracite stoker this year in my own home. It is a large, old house with a relatively new hot-air furnace. The furnace is amply large and in good condition. The heating was never satisfactory with hand firing, but with this automatic anthracite stoker with bin-feed and ash removal, etc., I have been very much amazed at the splendid heating results obtained.

One thing that surprised me, which I ought to have known, was the different sort of heat effects from an automatic stoker than from hand-firing. When I first came into this Society in 1919, we used to have arguments on the floor about different kinds of heating effects in boilers; that is, hot gas effects and radiation effects.

I was surprised to see the different results and different heat distributions in the hot-air ducts that I got with a stoker. I am not entirely able to tell you why, but it is very much better, a much more even distribution, and the automatic control is very satisfactory. Rooms that were difficult to heat before now heat perfectly. I think part of this is due to the fact that the chimney comes up in the northwest corner of the house, and there is a higher stack temperature and a higher chimney temperature on the walls of the two rooms that were most difficult to heat. The panel heating effect from that chimney has helped undoubtedly. Also, the lower floors are warmer due to a warmer basement.

Speaking of bin-feed, there is only one trouble I noticed with all bin-feed stokers, and that is the coal has an extremely steep angle at which it will stand, and the height of a bin that you can put in your cellar is more or less limited. I use No. 1 Buckwheat, and my bin holds 6 tons. I fixed it with a sloping bottom and thought I was going to have it feed the 6 tons in without attention. It wasn't very long,

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however, before I found a hole right down through the middle of the coal and the stoker ran out of fuel, so I had to drag coal in every once in a while.

In answer to Colonel Danielson's question about operation in mild weather, I am amazed at the little bit of a ball of dull red fuel lying in the retort that will keep the fire going. In addition to a room thermostat, there is a second control from stack temperature; it is not from a time device. It will operate just enough to reactivate the fire and produce 95 F in the stack, or something like that. Then it will stop again. In mild weather there is no over-heating of the house at all. It is surprising how little fuel it takes to keep the fire alive. In case the fire goes out from lack of fuel supply and the stack temperature falls too low, the stack control trips out and has to be reset by hand when the fire is re-started.

There is one thing that I noticed, and I do not know why it is; maybe somebody can tell me. If the fuel supply in the bin does get low and the coal screw is exposed, gas gets in the house. There does not seem to be any leakage in the firepot. I believe it comes out through the empty coal feed screw. Of course, as soon as the fire finally goes out, the gas stops; but there is a tendency to gas when the blower runs with an empty fuel screw.

On the whole, it has been very much of a surprise to me how satisfactorily my house has been heated by this little anthracite stoker. It has also been very satisfactory in heating a domestic water supply, except during very mild weather. I have arranged a water back and a by-pass so that if the storage tank becomes too hot, water flows direct to a large radiator up in the hall. That prevents any over-heating of water. In mild weather we do not get enough hot water for domestic use, and must have an auxiliary heater, but on the whole our stoker has been very satisfactory.

More work needs to be done on the design of bin feeds. I suggest some fabricated steel bins that can be easily bolted together in the cellar. The angle of slide can be made correct to feed all the coal in, and a two-ton bin would be very convenient and inexpensive. A stoker job is not complete until the bin feed is right. Any other scheme seems like a makeshift.

T. H. URDAHL: Looking back somewhat over a quarter of a century to my days as a twelve-year-old boy, I am singularly impressed with the fact, comparatively speaking, of the lack of improvement in our solid fuel heating facilities.

If we go back 200 years to the days of Benjamin Franklin, and read some of his writings, we will find that he was very much interested in the heating problem, and he made many suggestions for needed research at the time, which, apparently, are just being followed up today. In fact, our greatest development, apparently, has been taking place in the last six years; but I want to leave one word of caution. Our development seems to be going into the line of overgadgetizing. By that I mean using that very able servant, electricity, in burning our fuel. A portion of our population, in fact, a great portion live in the country. Electricity is being made available, but in the event of some national emergency, storm, war, flood, this facility would be gone.

No. 1119

PERFORMANCE OF STOKER-FIRED AND HAND-FIRED WARM-AIR FURNACES IN THE RESEARCH RESIDENCE

By A. P. KRATZ,* S. KONZO,** AND R. B. ENGDAHL,*** (*MEMBERS*), URBANA, ILL.

INTRODUCTION

Preliminary Statement

THE investigations in forced-air heating in the Research Residence at the University of Illinois during the period from 1932 to 1934 were confined to studies [1] of the characteristics of the forced-air heating system under actual service conditions. For these investigations the fuel used was anthracite. During the period from 1934 to 1937 the performance and operating characteristics were determined [2, 3] of a warm-air furnace equipped with a conversion oil-burning unit and of a warm-air furnace designed specifically for oil firing. During the heating season of 1937-1938, these investigations were extended to include the comparison of the performance and operating characteristics of a thermostatically-controlled hand-fired, warm-air furnace burning a high volatile bituminous coal, with those of the same furnace fired by means of a domestic stoker, of the underfeed type. In order to obtain comparable data no changes were made in the furnace itself, in the plant, in the volume of air circulated, nor in the settings of the thermostat when the stoker was substituted for hand firing. Furthermore, in each case, coal from the same mine and of the size usually recommended for the particular method of firing was used.

DESCRIPTION OF THE RESEARCH RESIDENCE AND HEATING EQUIPMENT

The Research Residence in Urbana, Ill., and the heating plant have been completely described in a previous publication.[4] The Residence is a three-

* Research Professor, Engineering Experiment Station, University of Illinois.
** Special Research Assistant Professor, Engineering Experiment Station, University of Illinois.
*** Research Assistant, Engineering Experiment Station, University of Illinois.

[1] University of Illinois, *Engineering Experiment Station Bulletin 266*, by A. P. Kratz and S. Konzo. 1934.

[2] Performance of Oil-Fired, Warm-Air Furnaces in the Research Residence, by A. P. Kratz and S. Konzo. (ASHVE TRANSACTIONS, Vol. 43, 1937, p. 215.)

[3] Study of Methods of Control and Types of Registers as Affecting Temperature Variations in the Research Residence, by A. P. Kratz and S. Konzo. (ASHVE TRANSACTIONS, Vol. 44, 1938, p. 309.)

[4] Loc. Cit. Note 1.

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story structure of standard frame construction, in which the walls are not insulated, and no weatherstripping is used at the windows nor at the doors. The wall section consists of lap siding, building paper, sheathing, 6-in. studding, wood lath, and plaster with rough sand finish. The windows are double hung, and during all of the tests reported in this paper they remained tightly locked.

The total space heated consisted of three rooms, a sun room, a breakfast nook, and a hallway on the first story; three rooms, a bathroom, and a hallway on the second story; and two rooms, a bathroom, and a hallway on the third story. The total volume of this heated space, from which the basement was excluded, was approximately 17,540 cu ft. The calculated heat losses were approximately 159,000 Btu per hour at an indoor-outdoor temperature differ-

TABLE 1—DIMENSIONS AND AREAS OF FURNACE USED

Grate diameter	23 in.
Grate area	2.88 sq ft
Firepot diameter	27 in.
Heating surface	
ash-pit	1.36 sq ft
firepot	8.10 sq ft
dome	18.87 sq ft
radiator	31.93 sq ft
Total	60.26 sq ft
Ratio of heating surface to grate area	20.9
Casing diameter	50 in.
Free area through casing	4.97 sq ft
Combustion space[a]	8.8 cu ft
Free area through over-fire damper	4.9 sq in.

[a] Combustion space is defined in this case as the space above the hearth level, including the dome, but not the feed neck.

ence of 80 F. The Residence is completely furnished, and during the heating season it was occupied by six people.

The heating plant consisted of a warm-air furnace used in connection with a forced-air heating system. Three cold-air returns were used and were connected into a cold-air box above the inlet to a centrifugal type of fan. The furnace was placed at the East end of the basement, and the warm-air registers were served from two main trunk systems.

The furnace used in all of the tests made with both the hand firing and stoker firing was thermostatically controlled, was of the cast-iron circular-radiator type, having a 27-in. firepot and 23-in. grate, and was equipped with a casing 50 in. in diameter. The areas of the heating surfaces and the dimensions of the furnace are presented in Table 1.

In order to install the mechanical stoker, which was inserted through the ash-pit door of the furnace, the grates were removed, and firebrick were arranged to form a flat hearth as shown in Fig. 1. The stoker was of the underfeed type, and coal was delivered from the hopper by means of a rotating screw to the retort, which was located in the middle of the hearth.

The hopper had a capacity of 300 lb. Both the rate of fuel input and the rate of air supplied to the tuyeres could be independently regulated. In the case of the tests reported in this paper, the rate of fuel input was maintained at 48.6 lb per hour and the rate of air input was adjusted to provide for satisfactory combustion conditions. In this case the CO_2 content of the flue gas was maintained at approximately 11 per cent when the stoker was in operation, and a slightly hazy atmosphere over the fuel bed was obtained. The rate of air input was not sufficient to blow particles of coal away from the fuel bed.

The rate of burning of the fuel is dependent on the rate of air supply to the fuel bed. It may be observed, however, that in the case of the

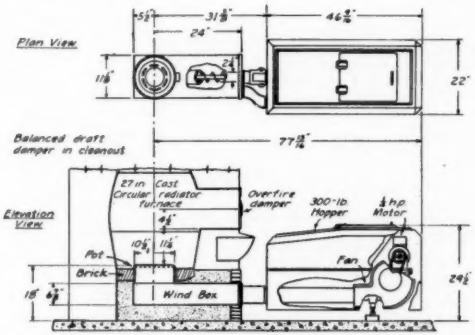

FIG. 1. DIAGRAM OF CONVERSION STOKER INSTALLATION IN FURNACE. RESEARCH RESIDENCE, SEASON 1937-38, SERIES 3-37 AND 4-37

domestic stoker of the underfeed type the rate of burning of the fuel is not necessarily the same as, and is usually less than, the rate of fuel input to the furnace. If the rate of burning of the fuel in an underfeed stoker were equal to the rate of fuel input, as it is on the chain grate stoker, then no excess fuel would be available in the firepot to maintain combustion during the off-periods. It is probable that with intermittent operation of the stoker, with a given rate of fuel input and depth of fuel bed, some leeway exists in the adjustment of the air volume which will result in satisfactory combustion conditions, and which will at the same time not materially change the level of the fuel bed. Hence, in the case of intermittent operation of the stoker in which some air is inevitably supplied to the fuel bed during the off-period, the rate of burning of the fuel should be such that:

1. Satisfactory combustion conditions are maintained
2. Sufficient coal remains in the fuel bed when the stoker operation is stopped to allow for combustion during the off-periods of the stoker
3. Sufficient heat is generated to handle the maximum demands of the house, and
4. The particles of coal are not blown away from the fuel bed.

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The fuel burned in these tests was a high-volatile bituminous coal obtained from Saline County, Ill. The characteristics of the fuel are shown in Table 2. In general the fuel used proved satisfactory for both hand firing and stoker firing.

TABLE 2—CHARACTERISTICS OF COALS

GENERAL DESIGNATION		
Size	Stove (2 in. x 1½ in.)	Stoker (1 in. x 10 mesh)
State	Illinois	Illinois
Vein	No. 5	No. 5
County	Saline	Saline
Type	Non-caking	Non-caking
Treatment	Washed, oil-treated	Washed, oil-treated
PROXIMATE ANALYSIS (As received) per cent:		
Moisture	3.50	8.50
Volatile matter	38.22	30.96
Fixed carbon	51.61	53.15
Ash	6.67	7.39
ULTIMATE ANALYSIS (Dry basis) per cent:		
Carbon	76.30	74.95
Hydrogen	4.90	4.73
Oxygen	8.28	8.94
Nitrogen	1.56	1.47
Sulphur	2.07	1.84
Ash	6.89	8.07
CALORIFIC VALUE, Btu per pound:		
Coal as received	13,041	12,107
Dry coal	13,514	13,232
Unit coal	14,670	14,550
Air required, pound per pound dry coal	10.3	10.0
Friability Index, D_w, per cent[a]	39.8	39.8
ASH CHARACTERISTICS, deg F.[b]		
Initial deformation	1,925	1,925
Softening temperature	2,000	2,000
Fluid temperature	2,193	2,193

[a] University of Illinois Engineering Experiment Station Bulletin No. 218.
[b] Average values from Illinous State Geological Survey for Saline County coal.

The centrifugal fan in the forced-air system delivered approximately 1675 cfm of air which was recirculated through the house. Six replaceable air filters of the viscous-coated fiber type, each 16 in. × 25 in., were installed on the inlet side of the fan.

The control of the heating plant was accomplished by means of a room thermostat operating to start and stop the stoker motor, or to open and close the ash-pit damper, and to start and stop the circulating fan as shown in

Fig. 2. This room thermostat was used in conjunction with two bonnet thermostats which served as high- and low-limit controls for the temperature of the air in the furnace bonnet. The settings for the thermostats are shown in Table 3.

This method of control [5] was highly satisfactory for previous installations in which anthracite and oil fuels were burned, and also proved to be equally satisfactory for both hand firing and stoker firing of bituminous coals. As

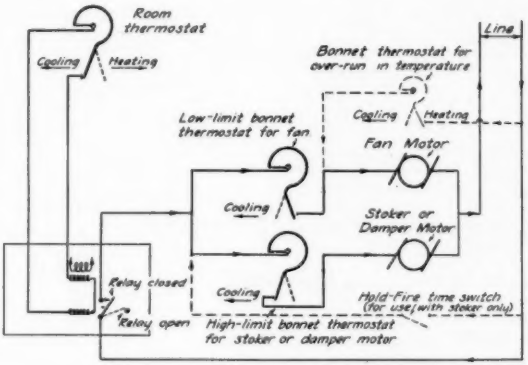

FIG. 2. SIMPLIFIED WIRING DIAGRAM FOR CONTROL OF FAN MOTOR AND STOKER MOTOR (OR DAMPER MOTOR). RESEARCH RESIDENCE INSTALLATION, SEASON 1937-38, SERIES 1-37, 3-37 AND 4-37

indicated by the broken lines in the upper right hand portion of Fig. 2, a contact was provided in the bonnet thermostat which closed the fan circuit when the bonnet temperatures exceeded, or over-ran, a temperature of 195 F. This same basic control system was used for the stoker-fired plant and, in addition, an auxiliary switch, designated as the hold-fire time switch, was included. This time-switch is indicated by the broken line in the lower part of Fig. 2. If the bonnet temperature were less than 125 F and if the stoker remained inoperative for 30 min, the time switch closed and the stoker was caused to operate for a two-minute period. This period of operation was sufficient to maintain the fire in the fuel bed, and, except during very mild days when there existed no demand for heat, did not cause any unsatisfactory overheating in the rooms in the house.

The room thermostat, which was of the heat-anticipating type, was located on an inside wall of the dining room at a height 60 in. from the floor, and was adjusted to maintain an average air temperature of approximately 72 F at the 60-in. level in all of the rooms of the Residence.

[5] Control Type IV described in the paper, Automatic Controls for Forced-Air Heating Systems, by S. Konzo and A. F. Hubbard, (ASHVE TRANSACTIONS, Vol. 40, 1934, p. 37.)

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METHOD OF CONDUCTING TESTS

The fan and the damper motor, or stoker, were automatically controlled to maintain an average temperature of 72 F in the rooms of the house both day and night.

Observations of weather, indoor room air temperatures, room relative humidities, and other incidental data were made daily at 7:00 a.m., 11:00 a.m., 4:00 p.m., and 10:00 p.m. Complete data were obtained for each 24-hour test

TABLE 3—APPROXIMATE THERMOSTAT SETTINGS, DEG F

	HAND FIRED	STOKER FIRED
Room thermostat setting	72	72
Lower-limit bonnet thermostat for fan motor:		
Make circuit[a]	125	125
Break circuit[a]	110	110
High limit bonnet thermostat for stoker or damper motor:		
Make circuit[a]	135	135
Break circuit[a]	150	150
Bonnet thermostat for over-run in temperature:		
Make circuit[a]	195	195
Break circuit	190	190
Hold-fire time-switch	...	2 min operation every 30 min

[a] These temperatures are not settings. They correspond to the actual air temperatures in the bonnet outlet as obtained from the temperature recorder in the North Trunk.

period on the fuel consumption, weight of ash and clinkers removed, the total integrated time of operation of the fan and of the stoker, the total electrical energy consumption of the fan and of the burner motors, and the total number of on-periods of both the circulating fan and stoker. Daily observations were made of the volume of air circulated, and the filters were cleaned with sufficient frequency to maintain the air volume constant. In addition, continuous records of temperatures and CO_2, such as are shown in the typical records in Figs. 3, 4, and 5, were obtained for each 24-hour period. Continuous records of the index of smoke density were also obtained by means of the apparatus shown in Fig. 6. Observations were made, as will be explained in detail later, on the time consumed in attending the furnace.

For each series of tests, data were obtained for a wide range of outdoor weather conditions, and significant values for each 24-hour period were plotted against the difference in temperature existing between the indoors and outdoors for the same period, as shown in Figs. 7 to 11 inclusive. The plotted points deviate to some extent from the curves representing the average of the observed data, and these deviations can be partly attributed to the wind and sun effects which cannot be represented on curves based on temperature difference alone.

In the first series of tests, which has been designated as Series 1-37, the furnace was fired by hand at 11 a.m., 4 p.m., 10 p.m., and 7 a.m. daily. The

fuel bed was poked, grates shaken, and ashes and clinkers removed once each day at 11 a.m. except in extremely mild weather when the fuel bed was disturbed only once every two or three days. Previous to each firing of the fuel the live coals and coke in the fuel bed were shoved back toward the rear of the firepot and the fresh charge of coal was fired in the front. The

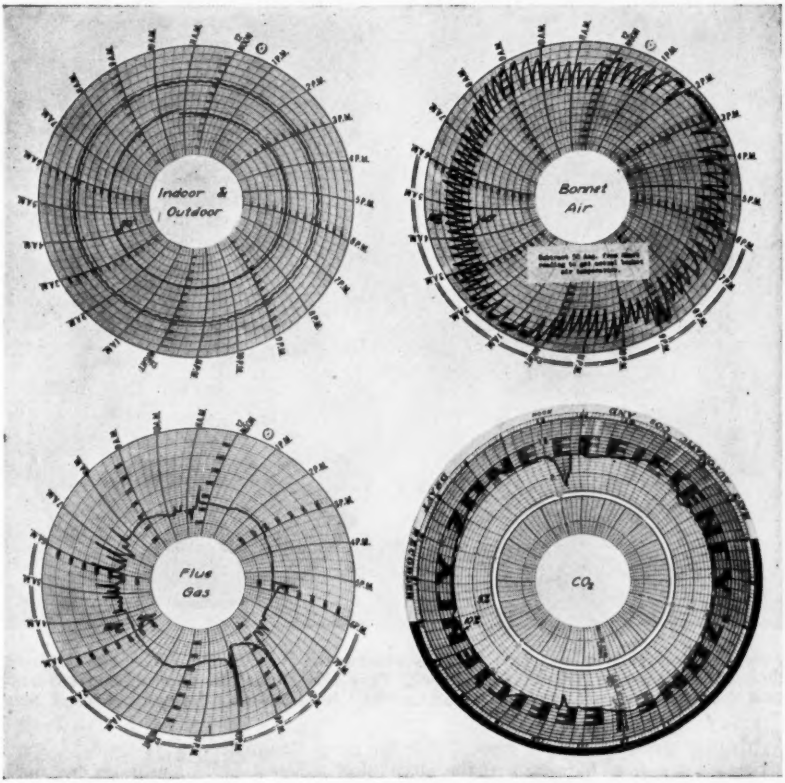

FIG. 3. TEMPERATURE AND CO_2 RECORDS FOR HAND-FIRED PLANT (OVER-FIRE DAMPER OPEN). SEASON 1937-38, SERIES 1-37, TEST NO. 1256. FORCED-AIR HEATING SYSTEM, CAST, CIRCULAR RADIATOR, 27-IN. FURNACE. BITUMINOUS COAL, 2 IN. X 3 IN. STOVE SIZE

over-fire damper in the firing door was left wide open at all times. Both the ashpit damper and the check damper in the smoke pipe were operated by means of the damper motor. With the fuel used, the rate of combustion was found to be extremely responsive to increases in draft. The method of firing employed and the attention given to the furnace may be considered as simulating normally good firing practice.

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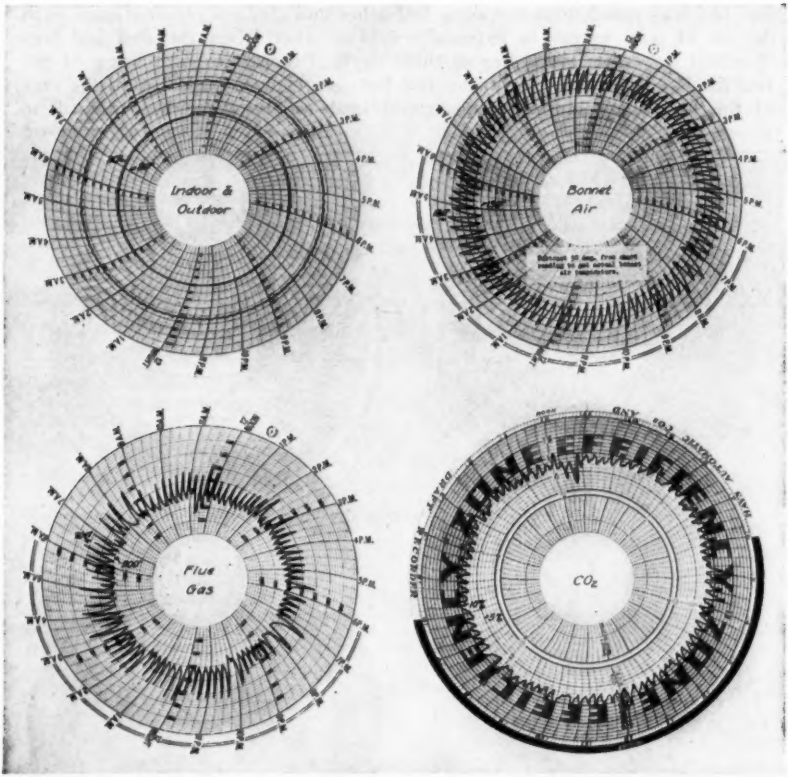

FIG. 4. TEMPERATURE AND CO_2 RECORDS FOR STOKER-FIRED PLANT (OVER-FIRE DAMPER OPEN). SEASON OF 1937-38, SERIES 3-37, TEST NO. 1265. FORCED-AIR HEATING SYSTEM, CAST, CIRCULAR RADIATOR, 27-IN. FURNACE. BITUMINOUS COAL, 1-IN. STOKER SIZE

In the second series of tests, which has been designated as Series 3-37, the furnace was fired by means of the mechanical stoker. At 11 a.m. each day the clinkers were removed, the fuel bed levelled, and the hopper filled with coal. During extremely mild weather no attention was given the fuel bed or hopper, except as required every two or three days. The over-fire damper in the firing door was left open at all times. A balanced check damper was installed in the clean-out of the chimney and was adjusted to maintain a draft in the smoke pipe of approximately 0.05 in. of water.

It was observed in the second series of tests that considerable burning accompanied by low CO_2 content of the flue gas occurred during the off-periods of the stoker. Since the burning that occurred during these relatively long periods was undoubtedly augmented by the air entering through the over-fire

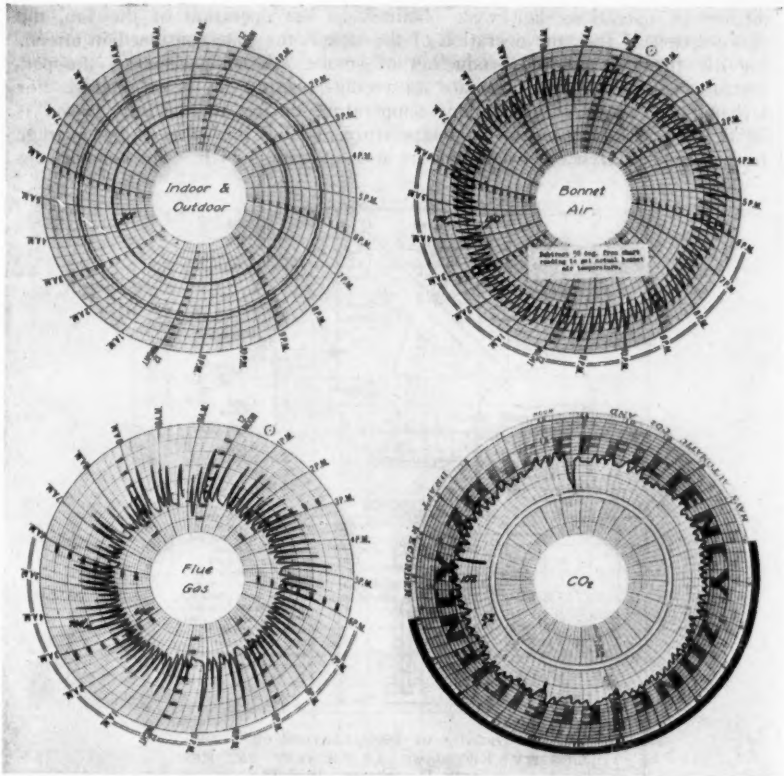

FIG. 5. TEMPERATURE AND CO_2 RECORDS FOR STOKER-FIRED PLANT (OVER-FIRE DAMPER CLOSED). SEASON OF 1937-38, SERIES 4-37 TEST NO. 1311. FORCED-AIR HEATING SYSTEM, CAST, CIRCULAR RADIATOR, 27-IN. FURNACE. BITUMINOUS COAL, 1-IN. STOKER SIZE

damper, the third series of tests, designated as Series 4-37, was run with the over-fire damper closed. In all other respects the method of operation was identical with that used in Series 3-37.

OPERATION OF CIRCULATING FAN

For these tests, which were made to study the comparative performance of the hand-fired and stoker-fired furnace plants under actual service conditions, both the circulating fan and coal stoker were operated intermittently by means of the automatic control system, so as to maintain uniform air temperatures in the house at all times. In determining the performance characteristics for each

method of operation, data were obtained on the operation of the fan, the consumption of fuel, the operation of the stoker, the time consumed in attending the furnace, and the production of smoke. A comparison of the performance of the heating plant for an average heating day is of interest. For Urbana, Ill., the average outdoor temperature during the heating season is 38 F. Hence for an indoor air temperature of 72 F the value of the average temperature difference between indoors and outdoors is 34 F. The comparisons

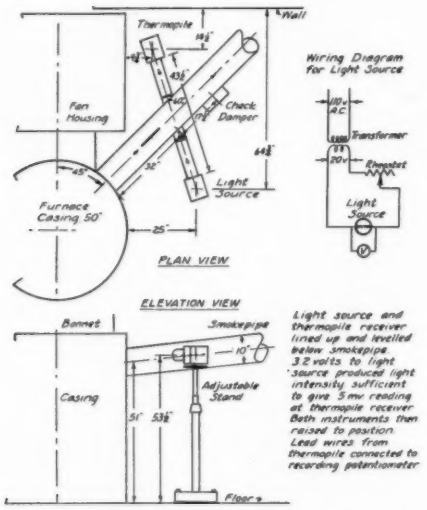
Thermopile
Wall
Fan Housing
Check Damper
Light Source
Furnace Casing 50"
PLAN VIEW
ELEVATION VIEW
Bonnet
Smokepipe
Casing
Adjustable Stand
Floor
Wiring Diagram for Light Source
Transformer
Rheostat
Light Source
Light source and thermopile receiver lined up and levelled below smokepipe. 3.2 volts to light source produced light intensity sufficient to give 5 mv reading at thermopile receiver. Both instruments then raised to position. Lead wires from thermopile connected to recording potentiometer

FIG. 6. DETAILS OF INSTALLATION OF SMOKE DENSITY RECORDING INSTRUMENT IN RESEARCH RESIDENCE, 1937-38

based on this temperature difference may be regarded as indicative of the results to be secured from the entire heating season.

As shown in Fig. 7, the total number of times the fan was operated, the total hours of fan operation, and the energy consumption of the fan motor per day were each smaller for the hand-fired plant than for the stoker-fired plant. It may be observed that for a temperature difference between indoors and outdoors of 34 F, the energy consumption of the fan was 2.1 and 2.8 kwhr for the hand-fired and stoker-fired plants respectively. This difference can be accounted for by the fact that with the hand-fired plant the mean of the actual bonnet air temperatures was higher than that for the stoker-fired plant, as shown in Fig. 8, and hence the demand of the room thermostat was satisfied by the delivery of warmer air for shorter periods of time.

It should be observed that the actual setting of the bonnet thermostat was the same for all tests, but on account of the larger amount of off-period burning

that occurred in the hand-fired plant when the ash-pit damper was closed, the whole range of bonnet air temperatures, and hence the mean bonnet air temperature, that was maintained was higher than that for the stoker-fired plant. The lower limit of the temperature range was high in the case of the hand-

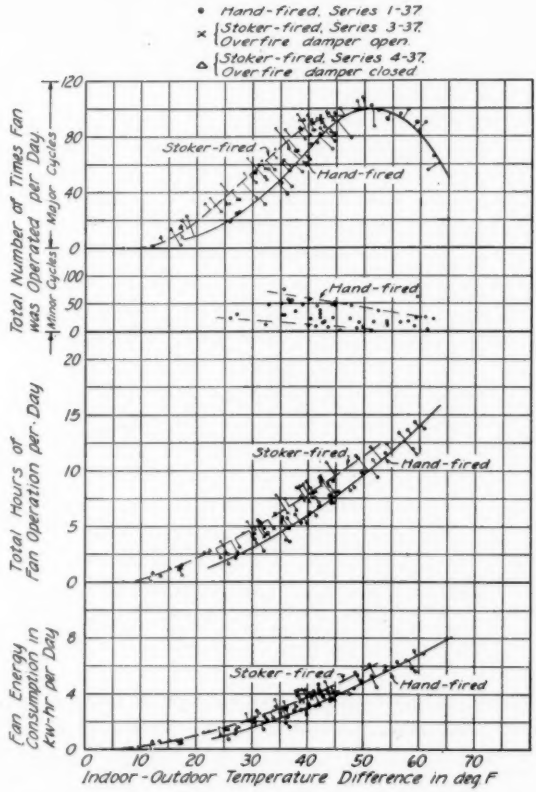

FIG. 7. OPERATING CHARACTERISTICS OF FAN. RESEARCH RESIDENCE TESTS. CAST, CIRCULAR RADIATOR, 27-IN. FURNACE. SEASON 1937-38, SERIES 1-37, 3-37 AND 4-37

fired plant because there was sufficient burning during the off-period, so that, sometimes, when the room thermostat demanded heat, the fan operated. However, the bonnet temperature did not decrease enough to actuate the bonnet thermostat in the damper motor line and thus open the ash-pit damper. When the bonnet air temperature became excessively high, the over-run contact shown in Fig. 2 closed; the fan operated irrespective of whether or not any demand

for heat existed, and the bonnet air temperatures were reduced below the upper limit temperature of 195 F. The bonnet air temperatures accompanying these periods of minor cycle operation are shown in the chart for bonnet air temperatures in Fig. 3. The frequency of these minor cycles in total number of operations per day is shown in the upper part of Fig. 7. On account of the slightly greater gravity action during the off-periods of the fan, resulting from the higher bonnet air temperatures, some amount of temperature over-run occurred in the second and third story rooms, even in moderate weather. The temperatures were, however, not unsatisfactory from the standpoint of comfort.

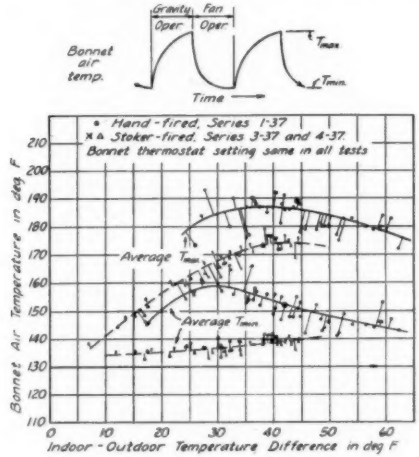

FIG. 8. OPERATING RANGE OF BONNET AIR TEMPERATURES. RESEARCH RESIDENCE DATA. CAST, CIRCULAR RADIATOR, 27-IN. FURNACE. SEASON 1937-38, SERIES 1-37, 3-37 AND 4-37

In the case of the stoker-fired plant the combustion rate was greatly diminished when the stoker ceased operation; the bonnet air temperatures were maintained within a lower range, as shown in Fig. 8; and no minor cycles of fan operation resulted. Hence, for moderate or severe weather conditions no overheating occurred in any part of the house. In extremely mild weather the maintenance of the fire by means of the hold-fire switch tended to produce some overheating in the house, particularly in the rooms on the second and third stories.

It is probable that if the combustion during the off-periods had occurred in the same manner for both the hand-fired and stoker-fired plant, the range of bonnet air temperatures would have been the same, and the performance of the fan would also have been the same in the two cases. In general, the slightly smaller energy consumption of the fan in the hand-fired plant was accomplished by a slight sacrifice in the exactness of the control of the temperatures in the bonnet and in the second and third stories of the Residence.

Hence, the reduction in energy consumption of the fan motor cannot be considered as an unqualified advantage.

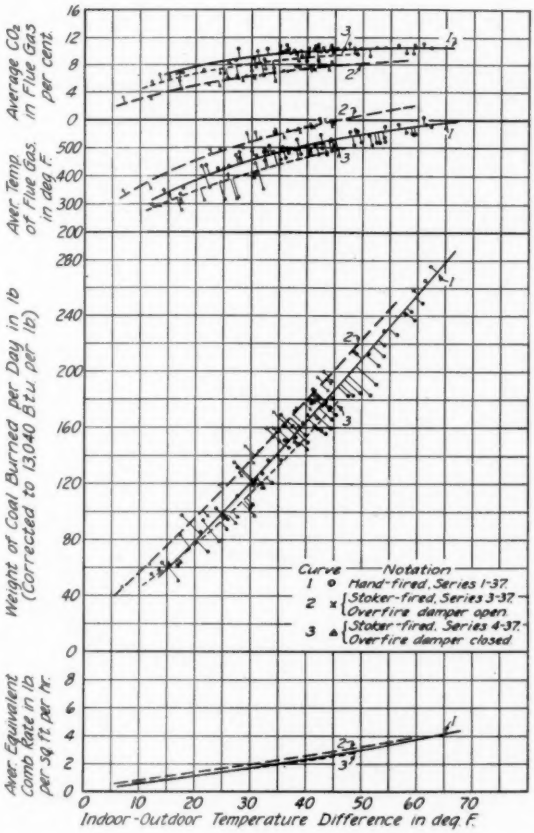
Average CO2 in Flue Gas per cent.
Aver Temp. of Flue Gas in deg. F.
Weight of Coal Burned per Day in lb (Corrected to 13,040 Btu per lb)
Aver Equivalent Comb Rate in lb per Sq ft per Hr.
Indoor-Outdoor Temperature Difference in deg. F.
Curve — Notation
1 ○ Hand-fired, Series 1-37
2 × {Stoker-fired, Series 3-37, Overfire damper open
3 △ {Stoker-fired, Series 4-37, Overfire damper closed

FIG. 9. PERFORMANCE CURVES FOR HAND-FIRED AND STOKER-FIRED FURNACES. RESEARCH RESIDENCE TESTS. CAST, CIRCULAR RADIATOR, 27-IN. FURNACE. SEASON 1937-38, SERIES 1-37, 3-37 AND 4-37

CONSUMPTION OF FUEL

Since small deviations occurred in the heating value of the different lots of coal used, the actual weight of coal fired in the stoker-fired plants was corrected to equivalent weight of coal, having a heating value of 13,040 Btu per pound,

and the corrected weights were plotted in the curves for fuel consumption shown in Fig. 9. It may be observed from these curves that the weight of coal burned in the hand-fired plant (curve No. 1) was less than that burned in the stoker-fired plant in which the over-fire damper was wide open (curve

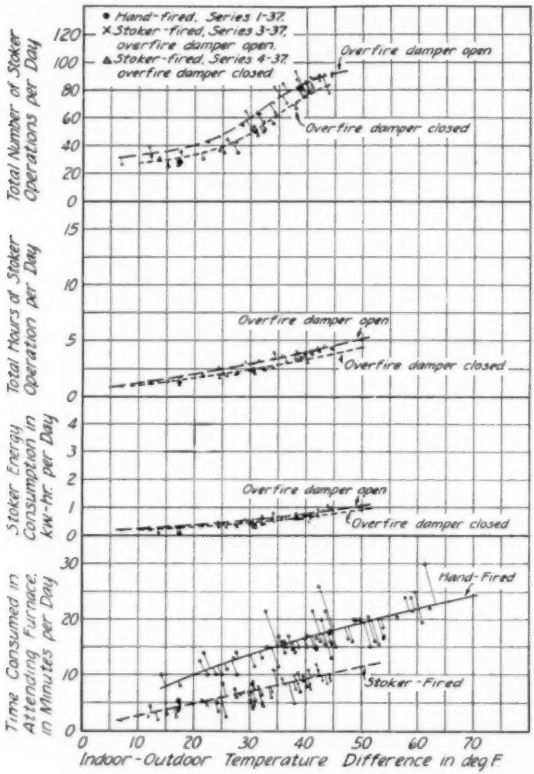
• Hand-fired, Series 1-37.
× Stoker-fired, Series 3-37, overfire damper open.
▲ Stoker-fired, Series 4-37, overfire damper closed
Total Number of Stoker Operations per Day
Overfire damper open
Overfire damper closed
Total Hours of Stoker Operation per Day
Overfire damper open
Overfire damper closed
Stoker Energy Consumption in kw-hr per Day
Overfire damper open
Overfire damper closed
Time Consumed in Attending Furnace, in Minutes per Day
Hand-Fired
Stoker-Fired
Indoor-Outdoor Temperature Difference in deg F

FIG. 10. OPERATING CHARACTERISTICS OF STOKER AND FURNACE. RESEARCH RESIDENCE TESTS. CAST, CIRCULAR RADIATOR, 27-IN. FURNACE. SEASON OF 1937-38, SERIES 1-37, 3-37 AND 4-37

No. 2), but was greater than that burned in the stocker-fired plant in which the over-fire damper was closed (curve No. 3). The values in Table 4, selected from Fig. 9, for an indoor-outdoor temperature difference of 34 F, show that the weights of coal burned per day were 138 lb for the hand-fired plant, 154 lb for the stoker-fired plant with over-fire damper open, and 128 lb for the stoker-fired plant with over-fire damper closed. Under local conditions a

price differential of approximately $1.25 per ton existed in favor of the coal used for the stoker-fired plant.

In the case of the hand-fired plant the fuel was burned comparatively uniformly, as indicated by the records for CO_2 and flue gas temperatures shown in

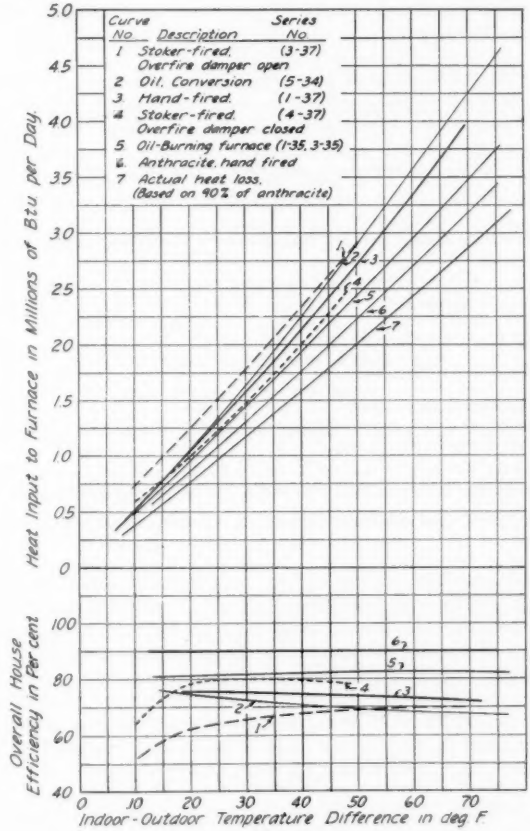
Curve No
Description
Series No
1 Stoker-fired, Overfire damper open (3-37)
2 Oil, Conversion (5-34)
3 Hand-fired. (1-37)
4 Stoker-fired. Overfire damper closed (4-37)
5 Oil-Burning furnace (1-35, 3-35)
6 Anthracite, hand fired
7 Actual heat loss, (Based on 90% of anthracite)
Heat Input to Furnace in Millions of Btu per Day.
Overall House Efficiency in Per cent
Indoor-Outdoor Temperature Difference in deg F

FIG. 11. HEAT INPUTS TO RESEARCH RESIDENCE. SEASONS 1934-38

Fig. 3. On the other hand, in both of the stoker-fired plants the evolution of heat was not as uniform as that for the hand-fired plant, as indicated by the records shown in Figs. 4 and 5. Furthermore, the burning that occurred during the off-periods of stoker operation was accompanied by a lower CO_2 content of the flue gas, particularly when the over-fire damper was open.

By proper adjustment of the air supply to the fuel bed during the periods of stoker operation, excellent conditions of combustion could be maintained. During the off-periods of the stoker, however, the fuel remaining in the fuel bed burned with the air supplied through the fuel bed by means of the existing natural draft and with the air supplied over the fuel fed through openings in the feed door. If a large amount of air were supplied from these two sources during the off-periods of the stoker, the amount of coal burned during the off-period would be large.

In the case of the hand-fired plant, the fuel bed contained a large amount of partly coked fuel which continued to evolve combustible gases during the off-period. These gases required a comparatively large amount of air for combustion, and the amount supplied by the over-fire damper was sufficient to promote good combustion of the gases, but did not result in a great amount of combustion of the fuel at the surface of the fuel bed. In the case of the

TABLE 4—COMBUSTION CHARACTERISTICS[a] OF THREE METHODS OF FIRING

FIRING METHOD	CURVE NO. IN FIG. 9	WEIGHT OF COAL BURNED, LB PER DAY	AVERAGE[b] CO_2 IN FLUE GAS, PER CENT	AVERAGE[b] TEMP. OF FLUE GAS, DEG F
Hand firing, over-fire damper open . . .	1	138	9.6	480
Stoker firing, over-fire damper open . . .	2	154	6.8	545
Stoker firing, over-fire damper closed . .	3	128	8.6	440

[a] Values selected from the performance curves in Fig. 9 for an indoor-outdoor temperature difference of 34 F.

[b] Values are average of those obtained during both the on-periods and off-periods of stoker operation.

stoker, the fuel remaining at the end of the operating period was well coked, and only a small amount of combustible gas was evolved during the off-period. Hence, with the over-fire damper open a considerably greater amount of air than that required for the combustion of the gases was drawn in, resulting in lower CO_2 content than that shown for the hand-fired plant. At the same time, however, the combustion of the fuel at the surface of the fuel bed was promoted and the evolution of heat from this source, together with the reduced heat transfer caused by the greater mass of air present, resulted in higher flue gas temperatures than those shown for the hand-fired plant.

When the over-fire damper was closed in the stoker-fired plant, the excess air was reduced, resulting in materially higher CO_2 values than those given by the same plant with the over-fire damper open. At the same time the restricted combustion at the surface of the fuel bed resulted in flue gas temperatures much lower than those obtained with the over-fire damper open in the same plant, and somewhat lower than those obtained in the hand-fired plant. The amount of leakage, however, was apparently sufficient, with the small amount of combustible gases evolved, to give more excess air than that obtained in the hand-fired plant, as evidenced by the lower CO_2 obtained in this case than that obtained in the hand-fired plant.

Since the total time of stoker operation per day, as indicated in the second group of curves from the top of Fig. 10, was ordinarily less than five hours, the off-period burning constituted an important part of the entire burning process. The result was that high or low CO_2 contents and temperatures of the flue gas occurring during these off-periods were reflected in corresponding high or low average CO_2 contents and flue gas temperatures as shown in Table 4. Since the flue gas loss is influenced by the combination of CO_2

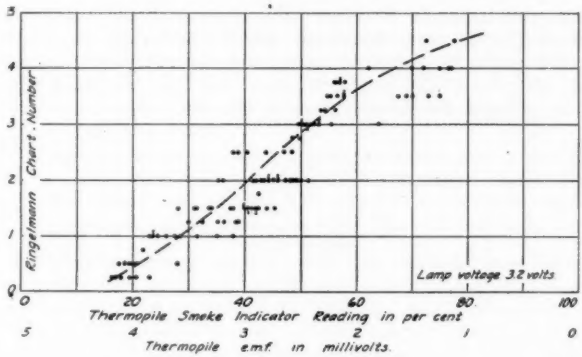
Ringelmann Chart Number
Lamp voltage 32 volts
Thermopile Smoke Indicator Reading in per cent
Thermopile e.m.f. in millivolts.

FIG. 12. CALIBRATION OF SMOKE DENSITY RECORDING INSTRUMENT

content and temperature of the flue gas, the higher fuel consumption of the stoker-fired furnace with the over-fire damper open, as compared with that of the hand-fired furnace, is consistent with the lower CO_2 content and higher flue gas temperature shown in Table 4. Also, the lower fuel consumption of the stoker-fired furnace with the over-fire damper closed, as compared with that of the hand-fired plant, is consistent with the slightly lower CO_2 content and the lower temperature of the flue gas shown in this case. This reduction in fuel consumption, however, was accompanied by the production of some smoke and soot, a discussion of which will be presented in a later section.

OPERATION OF THE STOKER

The number of stoker operations, the total hours of stoker operation, and the electrical energy consumption of the stoker motor per day are shown in Fig. 10. In each case the values obtained with the over-fire damper open were slightly greater than those obtained with the over-fire damper closed. For an indoor-outdoor temperature difference of 34 F the electrical energy consumption of the stoker motor was 0.6 kwhr per day with the over-fire damper open and 0.5 kwhr per day with it closed. The electrical energy consumption of the stoker motor per ton of coal fired was 7.8 kwhr for both methods of stoker operation. It is probable that with smaller rates of coal input to the furnace the values shown in Fig. 10 would have been larger.

TIME CONSUMED IN ATTENDING FURNACE

For the purpose of determining the time consumed in attending the furnace, measurements were made by means of a stop watch of the time consumed in shaking grates, removing ashes or clinkers, firing coal into the furnace or hopper, and leveling the fuel bed. The total time consumed per day is shown in the lower group of curves in Fig. 10. These values do not include the time necessary to walk from the first-story living room to the furnace and back. In the case of the Research Residence approximately 1¾ min per trip were required. Hence, if it is desired to account for this time, an addition of 4 x 1.75 or 7.0 min for the hand-fired plant and 1¾ min for the stoker-fired plant may be added to the values shown in Fig. 10.

The total time consumed in attending the furnace on an average day in which the indoor-outdoor temperature difference was 34 F was $15 + 7 = 22$ min for the hand-fired plant and $8 + 1\frac{3}{4} = 9\frac{3}{4}$ min for the stoker-fired plant, or a difference of 12¼ min per day. For a heating season consisting of 200 days, this difference would amount to $\frac{200 \times 12.25}{60}$ or approximately 41 hours.

OVER-ALL HOUSE EFFICIENCY

The significance of over-all house efficiency was discussed in a previous publication,[6] and further data on this subject were obtained on the stoker tests. In order to compare the fuel consumption with oil, anthracite, and bituminous coal, the fuel quantities shown in Fig. 9 were reduced to terms of heat input to the furnace, in millions of Btu per 24 hours, and were plotted as shown by curves Nos. 1, 3, and 4 in Fig. 11. The values for oil, (curves Nos. 2 and 5) and anthracite (curve No. 6) have been reproduced from a previous paper.[7] In all cases the operation of the heating plant was controlled by means of sensitive thermostatic controls.

From analysis of the flue gases when anthracite was used as a fuel, the heat lost in the gases escaping from the chimney, which represented heat furnished by the fuel that was not ultimately available for heating the house, was determined as approximately 10 per cent of the total heat input to the furnace. The actual heat loss from the house (curve No. 7) was then derived from the experimental results for heat input obtained with anthracite as a fuel (curve No. 6). By using these derived values of the actual heat loss from the house in connection with the fuel consumption curves for the hand-fired and stoker-fired plants, the over-all house efficiency, or the ratio of the heat loss from the house to the heat input to the furnace, could be calculated as shown in the lower part of Fig. 11. For an indoor-outdoor temperature difference of 34 F the over-all house efficiencies were 80 per cent with the stoker-fired plant

[6] Investigation of Warm-Air Furnaces and Heating Systems, Part IV, by A. C. Willard, A. P. Kratz, and V. S. Day, (University of Illinois, *Engineering Experiment Station Bulletin 189*, Chapter V).
[7] Loc. Cit. Note 2.

having over-fire damper closed, 75 per cent with the hand-fired plant, and 67 per cent with the stoker-fired plant having over-fire damper open. The value of 75 per cent obtained with the hand-fired plant using bituminous coal is in substantial agreement with results [8] previously obtained in the Research Residence.

The results obtained from these tests were to a great extent dependent upon the method of operation used. The conditions of hand firing which were maintained simulated normal intelligent firing of a thermostatically-controlled hand-fired plant. In the case of the hand-fired plant, it is probable that the fuel consumption would have been larger if lump coal, rather than sized coal adapted to hand firing, had been used; if hand control, rather than automatic thermostatic control, had been used; or if the coal had been thrown on top of the fire rather than to one side.

SMOKE DENSITY

The smoke density meter, consisting of a lamp and a thermopile, as shown in Fig. 6, used in connection with a recording potentiometer, provided a means for obtaining continuous records of the index of smoke density in the smoke pipe. The lamp voltage was adjusted, by means of a rheostat, so that when no smoke passed through the smoke pipe a maximum reading of five millivolts was obtained from the thermopile upon which the light beam was received. Smoke passing through the smoke pipe reduced the intensity of the light beam reaching the thermopile. When the light beam was completely intercepted the thermopile gave a reading of zero millivolts.

For the purpose of calibrating the instrument in terms of the more familiar Ringelmann numbers, the electromotive force generated by the thermopile was observed by means of the potentiometer, and at the same time the smoke emitted from the top of the chimney was compared with Ringelmann charts. The calibration curve for the instrument, with the check draft sealed in order to prevent air leakage between the instrument and the top of the chimney, is shown in Fig. 12. It may be noted that an indicator reading of 2.4 millivolts was obtained with smoke density corresponding to No. 3 Ringelmann. It is common practice for smoke ordinances to limit the production of smoke to six minutes of No. 3 smoke in an hour.

The index of smoke density, plotted against the time in hours after the coal was fired, is shown for the hand-fired furnace in the upper part of Fig. 13. The scale of Ringelmann numbers is given on the right hand side of the graph. It was observed that, when coal was added to the fuel bed at a time when low combustion rates were maintained, the comparatively low temperatures of the fuel bed resulted in correspondingly small rates of evolution of the gases and of formation of smoke. This condition is shown by the curve designated as the *upper range* in Fig. 3, and it may be observed that in this case the recommended limit of No. 3 smoke for a period of six minutes was never exceeded. However, when coal was added to a comparatively hot fuel bed, it was rapidly

[8] Loc. Cit. Note 6.

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volatilized and large quantities of smoke were evolved for a period of approximately 15 min after the coal was fired, as shown by the curve designated as the *lower range* in Fig. 13. Under these conditions the smoke given off by the fresh charge of coal exceeded the maximum limit of six minutes per hour of No. 3 Ringelmann density. In general, the amount and duration of the smoke produced by the hand-fired furnace were largely dependent upon the tempera-

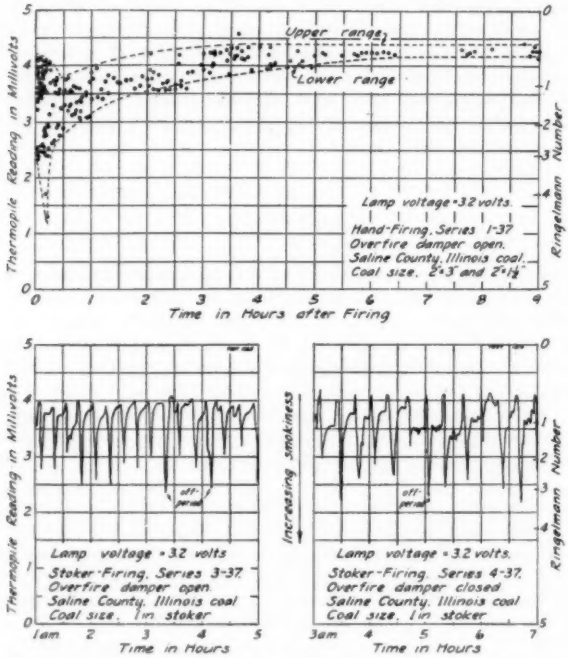

FIG. 13. SMOKE DENSITY RECORDS FOR HAND-FIRED AND STOKER-FIRED FURNACE. RESEARCH RESIDENCE DATA. SEASON 1937-38, SERIES 1-37, 3-37 AND 4-37

ture of the fuel bed at and immediately following the time of firing. After the gases had been driven off from the fresh charge of coal, and the fuel had been coked, the density of smoke was ordinarily less than No. 2 Ringelmann.

In both cases with stoker firing, as shown in the lower diagrams in Fig. 13, the density of smoke during the on-periods of the stoker was less than No. 1 Ringelmann and could be considered as practically negligible. During the period of approximately 30 seconds after the stoker was stopped the smoke density increased sharply to attain a maximum value of about No. 3 Ringelmann. It then decreased rapidly during the succeeding two minutes. For a few seconds in each hour a smoke density of No. 3 Ringelmann was obtained.

The smoke conditions accompanying the use of the stoker-fired furnace with the over-fire damper open could be considered as entirely satisfactory. However, with the over-fire damper closed, the combustion which occurred during the off-periods of the stoker was accompanied by some evolution of smoke and the formation of soot which coated the heating surface. It is probable that operation of the furnace for an entire season under these conditions would have resulted in some decrease in effectiveness of heat transmission and in some difficulties in maintaining proper draft. Hence, although the closing of the over-fire damper resulted in better fuel economy, this economy was obtained by a sacrifice of clean heating surfaces. Some amount of over-fire air was necessary in order to prevent the accumulation of soot. It is probable that the amount of over-fire air necessary would to a great extent be dependent upon the type of coal and the type of furnace and stoker used.

INCIDENTAL DATA

With the cooperation of the Illinois State Geological Survey, motion pictures of the fuel bed in the stoker-fired plant were made. Typical photographs showing various stages in the combustion process were made from enlargements of the original film and are shown in Fig. 14. Figs 14a and 14b show conditions common to both series of tests. Figs. 14c and 14d refer to conditions with the over-fire damper open, and Figs. 14e and 14f to those with the over-fire damper closed. None of these photographs shows any evidence of coke-tree formations. Furthermore, no difficulty was experienced from failure to form clinkers. Fig. 14a represents the fuel bed immediately after the stoker started, and shows active combustion starting in the center. The clarity of details indicates the entire absence of smoke. Fig. 14b was taken after the combustion had progressed for five minutes, and shows active and smokeless combustion occurring over the major part of the fuel bed. Fig. 14c shows that, immediately after the stopping of the stoker and with the over-fire damper open, the combustion in the fuel bed continued rather actively with air furnished through the over-fire damper, and that a slight amount of smoke was formed. Fig. 14d taken five minutes after the stoker was stopped shows that, with the over-fire air damper open, combustion in the fuel bed continued even during the off-period of the stoker. It may also be noted that the production of smoke had ceased by this time.

Fig. 14e shows that, immediately after stopping the stoker with the over-fire damper closed, combustion again continued in the fuel bed. However, in this case the reduction in the amount of over-fire air resulted in the formation of a considerable amount of smoke, and the evolution of smoke continued for five minutes after the stoker was stopped, as shown by Fig. 14f. Even in this case combustion continued during the off-period of the stoker, but not as actively as it did with the over-fire damper open.

An inspection of the radiator, the smoke pipe, and the chimney clean-out indicated that in the case of the stoker-fired plant considerable amounts of fly ash were deposited. The ash per pound of coal fired (theoretical ash) was 7.39 per cent; whereas the actual weight of all the ash and clinkers,

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(*a*) Beginning of stoker operation

(*b*) After five minutes of stoker operation

(*c*) Beginning of off-period, over-fire damper open

(*e*) Beginning of off-period, over-fire damper closed

(*d*) Five minutes after stopping stoker, over-fire damper open

(*f*) Five minutes after stopping stoker, over-fire damper closed

FIG. 14. PHOTOGRAPHS OF STOKER FUEL BED

together with any unburned combustibles, removed from the fuel bed was 7.00 per cent by weight of the coal fired. A chemical analysis of the ash and clinkers removed from the furnace indicated that only a trace of unburned combustibles (less than 0.05 per cent) remained in the ash. Hence, the difference of 0.39 per cent of the weight of coal fired may be considered as an index of the amount of ash, in the form of fly ash, that was not removed through the feed door. This represents an amount equivalent to 0.39 lb of fly ash per 100 lb of coal fired, or 7.8 lb per ton of coal fired. In the case of the hand-fired plant the difference between the weight of the ash and refuse and the theoretical weight of ash was approximately five per cent. Since no appreciable amount of fly ash was in evidence, this five per cent may be regarded as combustible lost in ash.

These preliminary studies with a stoker-fired furnace have indicated that the results obtained are dependent upon the adjustments made in the operating conditions. Further tests are contemplated, particularly with reference to the effect of varying the rate of coal input and the rate of burning on the over-all plant performance.

SUMMARY AND CONCLUSIONS

The following summary and conclusions may be considered as applying to the Research Residence and the conditions under which the tests were conducted.

(1) The method of control used, consisting of a room thermostat and a bonnet thermostat which operated both the circulating fan motor and the stoker motor, was found to be highly satisfactory for a forced-air heating plant fired by an underfeed stoker.

(2) With a given indoor-outdoor temperature difference, the weight of coal burned in the hand-fired plant was less than that burned in the stoker-fired plant in which the over-fire damper was wide open, but was greater than that burned in the stoker-fired plant in which the over-fire damper was closed.

(3) The burning that occurred during the off-periods of stoker operation constituted a relatively important part of the entire burning process.

(4) For a coal input rate of 48.6 lb per hour, the electrical energy consumption of the stoker motor was 7.8 kwhr per ton of coal fired.

(5) The total time consumed in attending the furnace on an average day was 22 min for the hand-fired plant and 9¾ min for the stoker-fired plant.

(6) During the period of approximately 30 seconds after the stoker was stopped, the smoke density increased sharply to attain a maximum value of about No. 3 Ringelmann and then decreased rapidly during the succeeding two minutes.

(7) With the over-fire damper closed the combustion which occurred during the off-periods of the stoker was accompanied by some evolution of smoke and by the formation of considerable amounts of soot.

(8) In stoker-fired furnaces it is advisable to provide accessible clean-out doors for the removal of any fly ash which may accumulate in the gas passages.

ACKNOWLEDGMENTS

The results presented in this paper were obtained in connection with the investigation of warm-air furnaces and heating systems in the Research Resi-

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dence,[9] at the University of Illinois, conducted by the Engineering Experiment Station of which M. L. Enger, Dean of the College of Engineering, is the director, and in the Department of Mechanical Engineering of which O. A. Leutwiler, Professor of Mechanical Engineering Design, is the head. This investigation is a cooperative project sponsored jointly by the *National Warm Air Heating and Air Conditioning Association* and the Engineering Experiment Station. These results will ultimately comprise part of a bulletin of the Engineering Experiment Station. Acknowledgments are also due to Dr. Louis C. McCabe, and Dr. Harold B. Willman, Associate Geologists of the Illinois State Geological Survey, for their assistance in making motion pictures of the fuel bed. The assistance of the various manufacturers who cooperated by furnishing instruments and equipment is gratefully acknowledged.

DISCUSSION

C. J. SCANLAN[10] (WRITTEN): The introduction of the under-feed stoker for use in domestic heating plants was accompanied by many claims for greater economy. Some were based on a reduction in coal consumption, while others were based on the use of cheap grades of coal.

During the last two or three years both of these claims have been proved false, in the majority of cases. Stoker users have come to the realization that in order to secure the maximum operating convenience it is necessary to burn a coal having the required specifications for under-feed firing. Further, they now know after several winters' experience that they burn as many tons of coal per season as they did when hand firing their heating plant. However, in many cases the stoker coal was from one to two dollars lower in price than the coal used for hand firing.

The mechanical features of design have been perfected to a high degree, but apparently less attention has been given to the principles of combustion.

The combustion of coal is so common-place that the process is taken for granted. Combustion is the chemical process of combining oxygen with carbon and hydrogen. The rate of burning is effected by the weight of the various elements taking part in the process, as well as the temperature at which it occurs.

According to the law of chemical mass action, a chemical reaction proceeds in one direction until equilibrium is established and then stops; and when the system is in equilibrium there is for a given temperature a certain constant relation between the amounts of the components entering into the reaction.

More accurate knowledge about the processes which take place in the fuel bed has been obtained in the fuel bed of the gas producer than from studies of the combustion of solid fuel in heating and power boilers.

From the experiments of Herr Ernest carbon begins to combine with oxygen at 752 F and continues with increasing quantities until the temperature of 1292 F is reached at which point a maximum quantity of carbon dioxide is produced, providing of course that an ample supply of oxygen is present. An excess of oxygen does not affect the reaction.

As the temperature continues to rise, the quantity of carbon dioxide diminishes and the quantity of carbon monoxide increases, until a temperature of 1823 F is reached when carbon monoxide is all that can be produced regardless of the excess oxygen available.

[9] The Research Residence in Urbana, Ill. was built, furnished and completely equipped specifically for research work in warm-air heating by the *National Warm Air Heating and Air Conditioning Association* in December, 1924.

[10] M.E., Bloomington, Ill.

Further, the experiments of Dr. C. S. Hudson demonstrated that the amount of carbon monoxide formed at a given temperature depends largely on the time of contact or in other words on the rate of flow of the gas through the fuel bed.

Also from the work of J. K. Clement reported in Bulletin No. 30, University of Illinois Experiment Station, it will be noted that with increasing temperature there is a rapid increase in the percentage of carbon monoxide obtained with any given rate of gas flow; and that with increasing gas velocity at low temperatures the percentage of carbon monoxide formed falls off rapidly, at higher temperatures very slowly.

Quoting from the Bureau of Mines, Technical Paper No. 139, "The burning fuel bed acts principally as a gas producer and a gas retort; in it the solid fuel is gasified by the partial combustion of the fixed carbon and by the distillation of the volatile matter. The gases rising from the fuel bed contain at times as much as 32 per cent combustible, and no free oxygen to burn it. Hence, to burn these gases, air must be admitted over the fuel bed and mixed with the combustible, and the mixture burned in the combustion space.

"Of the air necessary for complete combustion of the fuel only about one-half can be supplied through the grate; the other half must be supplied above the fuel bed. If attempt is made to force more than this amount of air through the grate, only the rate of gasification of the coal in the fuel bed is increased, while the degree of completeness of combustion as indicated by the composition of the gases leaving the fuel bed remains the same. The combustion can be completed only in the combustion space by adding enough air over the fuel bed and mixing it with the combustible. In other words, the processes in the fuel bed determine the rate and the processes in the combustion space the completeness of the combustion. In a general way, it can be said that in burning coal one-half of the combustion takes place in the fuel bed and one-half in the combustion space.

"The preceding statement holds good for bituminous coal, anthracite, and coke, burned at ordinary and high rates of combustion. For low rates of combustion such as are frequently used in house-heating furnaces the process of combustion in the fuel bed is somewhat more complete and the gases rising from the bed contain smaller percentages of combustible gases."

Ordinary combustion rates are approximately 20 lb of coal per square foot per hour and high combustion rates are above that figure as applied to the above paragraph. The low rates referred to are below 10 lb per hour per square foot.

The same report shows, "The oxygen flowing up through the grate is all used as it passes through the first 3 or 4 in. of the fuel bed. The products of combustion in this part of the fuel bed are largely carbon dioxide, and a smaller percentage of carbon monoxide. At a height of 3 or 4 in. from the grate the carbon dioxide content of the gases reaches a maximum of 12 to 18 per cent. As the carbon dioxide passes through the upper layers of the fuel bed it is reduced by contact with the hot coal to carbon monoxide, which with the combustible gases distilled near the surface of the fuel bed constitutes 16 to 30 per cent of the gases rising from the fuel bed."

These tests as reported indicate the necessity of a primary and a secondary combustion with solid fuels.

However, this paper by Prof. A. P. Kratz et al does not agree with the previously mentioned established facts. Table 4 shows a higher efficiency for a condition with the over-fire damper closed than with it open. The conclusions do not represent the facts.

Let us look at the facts as reported. The stoker fed coal at the rate of 48.6 lb per hour, which required that the stoker operate only 3 hours or less for a 34 F

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difference indoor to outdoor temperatures. With hand firing and the same temperature differences approximately 6 lb of coal would have been required per hour. If the coal were burned at the same rate as it was fed the combustion rate per square foot of retort area would be approximately 60 lb. It was necessary for the stoker to feed enough coal so that combustion could be maintained during the long off periods.

Apparently the curves of Fig. 9 represent a composite of the off and on conditions of the fire. This is illogical, because the two conditions have very little in common. Therefore, from Figs. 4 and 5, which gives the only operating data of the report, the following values were obtained:

Operating Condition	Flue Gas Temperature Deg F	CO_2 Per cent	Excess Air Per cent
Over-fire Damper Open [a]			
Stoker *ON*	650	11.75	60
Stoker *OFF*	475	7.50	130
Over-fire Damper Closed			
Stoker *ON*	708	9.75	81
Stoker *OFF*	433	7.15	140

[a] Tests nos. 1256 and 1311.

Combustion conditions with the over-fire damper open were better than that obtained by hand firing, whereas with the damper closed the combustion conditions were approximately the same as with hand firing. It is also clear that there are two combustion conditions to be considered. First, the conditions existing when the stoker operates, and second, the condition when the stoker is not in operation. The *OFF* period is more important than the *ON* because it affects both conditions. High rates of feed and regulation of the air pressure in accordance with the resistance of the fuel bed have been developed to overcome the coking of a fuel bed or in some cases fusing of the coal during the off period. Excess air at high pressures are then necessary to prevent the filling of the combustion space with coal. Also, because of the high rates of coal feed, thick fires have been maintained, which require large amounts of excess air. The proper formation of clinker has also been difficult unless strong air blasts are used.

Inspections made of stokers in operation in the field show without exception very strong air blasts and large quantities of excess air.

To improve stoker operation the coal feed must be reduced from that now used by most stoker manufacturers. The stoker should not feed coal at more than a 10 per cent increase above the coal required for hand firing in the coldest weather. Thus, for the Tests on the Research Residence the coal feed should have been approximately 18 lb per hour instead of the 48.6 lb that was used. With lower feed rates thinner fires will be obtained with the result that lower air pressures may be used. Thick fires are inefficient. A normal fire should not exceed 10 or 12 in. in thickness measured from the bottom row of tuyeres. This will permit the top row of tuyeres to deliver the air through the thin fire bed into the combustion space that is required for secondary combustion.

Constant air pressure devices serve no useful purpose in the control of combustion conditions and should not be used.

However, the quantity of air must be reduced to an amount that will not burn the coal as fast as the stoker feeds it, in order that a residue of coal will be available to

maintain the fire during the *OFF* period. With this air adjustment, in the majority of cases, insufficient air will be supplied to prevent coking or fusing of the coal, which will produce a fuel bed condition preventing good combustion conditions when the stoker starts for its next period of operation.

The air quantity requires accurate adjustment both during the *ON* and the *OFF* periods, if efficiencies are to be increased. Further, maintaining combustion during the *OFF* period will provide a more uniform supply of heat to the system, which will produce a smaller variation in room temperature.

Most stoker manufacturers have designed their stokers so a minimum amount of air is admitted to the fuel bed when the stoker stops, and in some cases an effort is made to permit no air during this period. There are no engineering data available to support this idea, whereas, there is ample evidence showing the necessity of adjusting the air supply during the *OFF* period.

A. J. JOHNSON: I hesitate to mention any practical experiences that I might have had in the field, for fear they would be construed as comparing them with the invaluable and incomparable work that the University of Illinois is doing.

However, Lee P. Hynes made a remark[11] that hit very close to home. He stated that he was getting comfort that he had never had before with an automatic stoker, and that he could not explain it. That checks with the experiences we have had and with a number of experiences brought to our attention. We have never been able to quite measure the difference between the comfort of a continuously fired plant and the comfort of an intermittent type.

For example, in my home I installed a gas furnace for six weeks. I then installed an oil burner, and that was followed, after a second six weeks, with an automatic stoker. We had instruments throughout the house, and made a very careful recording of everything that was pertinent. In spite of the fact that the instruments did show a slight advantage in favor of the stoker, my general observations were that the stoker was considerably ahead either of oil or gas in actual comfort; much more so, in fact, than was actually measured. I think that that can now be very definitely traced to the difference between the measurement of mere air temperature in the room and measurement of the comfort as tied in with air temperature, plus radiation to the cold wall.

In following that through, I discontinued the use of the thermostat at night, and started to keep the house at the same temperature night and day. Instantly I obtained the desired result. There was no question but that there was a marked difference between the comfort of that house with the temperature the same for 24 hours, as compared with the temperature lowered during the night, so that in the morning the walls were cold.

If I may offer a suggestion to Professor Konzo, it is that we all put our heads together and see if in future work of this type we cannot actually measure the thing in which we are interested; that is, the comfort of the house as, may I repeat, reflected by the radiation, plus the temperature, rather than what we now consider to be more or less of a half-reading, and that is the air temperature alone.

PROFESSOR A. P. KRATZ: The only question that was asked was in reference to comfort. I might point out in this connection that we made studies and presented them in a previous[12] paper and we found that the thing Mr. Johnson mentioned is true; that there was a distinct difference in comfort with the different types, controls, and maybe with different fuels. It puzzled us. Ordinary routine observations did not show us why it was. We finally traced the reason, but we had to do it by

[11] See Discussion on Small Stokers, p. 295.

[12] Study of Methods of Control and Types of Registers as Affecting Temperature Variations in the Research Residence, by A. P. Kratz and S. Konzo. (ASHVE TRANSACTIONS, Vol. 44, 1938, p. 309).

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making a very elaborate series of measurements with thermocouples at the floor at various points. We found that the comfort was largely a question of the relation between the on-cycles and the off-cycles and what went on between the on-cycles and the off-cycles, particularly in the region near the floor. If you operate the plant with a stand-by at night and let the house cool down, it is very probable that radiation from the walls may enter into the picture, but we operate our plant to maintain practically uniform temperatures during the 24 hours. While there is undoubtedly some effect of radiation from the walls under different conditions, I think probably that that is a minor effect.

We did do some work, both with oil and with gas, by lowering the house temperature at night down to about 60 degrees. In those tests we found that in the morning the radiation from the cold walls and cold floors, et cetera, did play a rather important part in how long it took to establish a comfortable condition the next day. I rather doubt whether that has very much bearing on a stoker test.

I would like further to add, as a matter of record, that in the case of our tests on the stoker, while the feed rate was 48.6 lb per hour during the on-periods of the stoker, the actual burning rate, as determined by the adjustment of the air rate was approximately 18 lb per hour.

AUTHORS' CLOSURE: Mr. Scanlan in his discussion refers to certain conclusions concerning combustion developed by Henry Kreisinger and his associates from work done in the U. S. Bureau of Mines. These conclusions are incontrovertible as applied to uniform fuel beds in which no air can pass through the bed and escape at the surface without coming into contact with incandescent fuel. This condition does not exist in the case of the small domestic stoker, particularly in the one used for these tests. Even with a high rate of coal feed the incandescent fuel does not fill the firepot, but forms a rim of clinkers, loose ash, and coke particles around the outside of the retort which offers a more or less free passage for air from the tuyeres in the outer surface of the retort to gain access to the surface of the fuel bed without passing though incandescent fuel. Hence the apparent over-fire is not all furnished through the over-fire damper, part of it being supplied by the tuyeres in the outer surface of the retort. The relative proportions being supplied in these two ways varies from time to time depending on the condition of the fire. For this reason the theories involved in the combustion in a uniform fuel bed do not apply, and attempts to draw conclusions from one or two instantaneous readings of CO_2 and flue gas temperatures may be misleading. The curves in Fig. 9 based on composite results represent efficiencies reflecting what the consumer will have to pay for fuel during a season and hence furnish the only reliable criterion from which conclusions may be drawn.

In the case of these tests the actual combustion rate was approximately 18 lb per hour as compared with the feed rate of 48.6 lb per hour. Mr. Scanlan states that the feed rate should not exceed the combustion rate by more than 10 per cent but offers no data in support of this. The experience gained on these tests proved that coal could be fed at 2.7 times the combustion rate without difficulty. Furthermore the high over-all house efficiency obtained indicated that no sacrifice in efficiency was made. Subsequent tests have further proved that with a feed rate as low as only 10 per cent in excess of the combustion rate the burning was less uniform, holes developed in the fuel bed, the flames were more in the nature of a torch, and no gain in efficiency was obtained. It should be noted that the stoker used on these tests had no cut-off damper in the air tube, and hence the fuel bed and blower offered the only resistance to the passage of air into the fire during the off periods of the stoker.

No. 1120

SMOKE ABATEMENT—WHERE TO DRAW THE LINE

By H. B. Meller,* Pittsburgh, Pa.

IF AIR pollution abatement is to make the advances demanded in cities, it seems clear that ordinances and methods of enforcement need to be scrutinized. Excellent work has been done by smoke-ordinance enforcement officials and their staffs, working under the conditions prescribed by law, in many localities. The demand is for more law enforcement in abating smoke nuisance and more of this mandatory regulation will be forthcoming in various communities. The responsibility rests with those familiar with the difficulties to see that the development is sound and along logical lines.

Current Pittsburgh Survey

Before discussing provisions of ordinances and possible methods of securing better results from enforcement, mention should be made about some of the information now being collected in Pittsburgh through a WPA grant. While there are many phases of this project, special interest at the moment is in the amount of dirt in the city and the sources from which it comes—in other words, soot fall plus visual observation of stacks.

For these purposes the city is divided into 100 districts, in each of which is a station for collection of sootfall by the method usually followed. There are 50 observers, each of whom works approximately 120 hours a month (6 hours a day for 20 days, part of the time from early morning until noon, and part of the time from noon until evening). Thus each man covers two of the sootfall districts, reporting source, density and duration of any smoke observed. Naturally, the observer cannot at one time see all of the stacks in a single territory, but this method of analyzation gives a fair coverage.

* Managing Director, Air Hygiene Foundation of America, Inc.; Head, Air Pollution Investigation, Mellon Institute.

Presented at the 45th Annual Meeting of the American Society of Heating and Ventilating Engineers, Pittsburgh, Pa., January, 1939.

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The data compiled at this time cover only the period from April through November, 1938. They are not to be considered conclusive but merely as illustrative. However, as investigators are desirous of analyzing results as they are tabulated, some interesting comparisons will be shown herewith. Bear in mind that additional data collected during the current year may change the picture materially.

The average sootfall shows the usual seasonal difference, varying from 85.1 tons per square mile in April to 39.6 tons per square mile in July, and back to 78.8 per square mile in October. Later months, November through March, will be higher.

The effect of topography, which has been referred to in previous papers, is interesting. The sootfall collection stations were divided between hill and valley locations. Averages for the hill stations are consistently about two-thirds as high as those for the valley stations.

From the reports of the observers an estimate of the volume of smoke should be possible later. This is highly important, since it takes as long for a smoke inspector to observe a small stack as a large one. At this stage of the study, however, the records show density and duration of each smoke and the stack from which it was emitted. Therefore in this paper reference will only be made to the *number of smokes.*

For the period under discussion, 66 per cent of all smokes reported were light in density (No. 1 Ringelmann Chart, or less), 21 per cent medium (No. 2) and 13 per cent dense (No. 3 or heavier).

Fuel-burning plants are classified as heavy industrial; light industrial; commercial, including heating plants in office buildings, stores, apartment buildings, etc.; and domestic. In the classes regulated by city ordinances are approximately 5000 stacks. In the city there are 154,000 dwelling units, in 119,000 of which the fuel used is coal; none of these is subject to smoke regulation.

Considering dense smoke only, a comparison of the various classes of plants is as given:

In March, for example—a high month for total smoke—of 21,439 cases reported, 2 per cent were in heavy industry, 6 per cent in light industry, 20 per cent commercial and regulated heating plants, and 72 per cent were domestic.

Contrast that with July. Of 5270 cases reported (and supervision over industrial plants was concentrated during the summer months), 18 per cent were in heavy industry, 54 per cent in light industry, 26 per cent commercial, and 2 per cent domestic (incinerators).

In November there were again 14,238 cases—4 per cent in heavy industry, 7 per cent in light industry, 18 per cent commercial and 71 per cent domestic.

The indications are that roughly 80 per cent of the winter smoke from buildings in the commercial class was from heating plants.

Conclusions are not to be drawn from these very incomplete data. Upon completion of the survey, this and other phases of the air pollution problem will be carefully analyzed. In this the assistance of selected groups of physicians, engineers and other specialists will be sought.

ORDINANCES AND ENFORCEMENT

What do the few data herein presented indicate? The indications are that the smoke abatement staffs of the cities are too small to maintain observational and inspectional supervision over their territories—which has always been known of course; and that major changes are needed in ordinances and in methods of enforcement. The latter item is especially true with regard to low pressure plants, including the enormous number of domestic boilers and furnaces.

VISIBLE SMOKE

A *Model Smoke Abatement Ordinance* prepared by the *Smoke Prevention Association* contains this section: "The emission of dense smoke within the city from the smoke stack of any locomotive, steamboat, steam tug, steam roller, steam derrick, steam pile-driver, tar kettle or other similar machine or contrivance, or from any open fire or from the smoke stack or chimney of any building or premises, excepting for a period of six minutes in any one hour during which the firebox is being cleaned out or a new fire built therein, is hereby declared to be a nuisance and may be summarily abated. . . ." Many industrial cities are now operating under such provision—no dense smoke except as specified, and no exemption for any class.

Probably this is as close as the limit should be drawn at this time, although fires *can* be cleaned and new fires *can* be built with little or no dense smoke. However, the belief is that an operation which can be carried on normally without dense smoke can also be carried on with the production of a limited amount of No. 2 smoke—for example 5 or 6 min at any one period.

For the present consider the matter of supervision, with inspectional forces in the large industrial centers in ratios (inspectors to population) of 1: 60,000 to 1: 300,000. Obviously the present methods of observation—inspection—instruction are not adequate, and a system should be devised which will result in satisfactory over-all control with a limited official personnel.

FLY-ASH

The control of fly-ash is now considered a necessary part of the smoke abatement program. The term has come to mean ash plus cinders plus coke—in other words, visible smoke and fly-ash together include all solid products of combustion. Fly-ash is prohibited in 33 of 77 cities whose ordinances were analyzed by the *Smoke Prevention Association*. Eighteen of these apparently have no provision for enforcement; nine require only screens at the top of the stack; only six appear to have definite requirements covering the installation of dust separating equipment. As it is known that emission from domestic and other small chimneys is relatively high in tar and carbon particles and relatively low in ash, it is also known that the reverse is true of the larger stacks with their better combustion and much higher gas velocities.

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The amount of fly-ash which should be permitted has not, as far as is known, been determined. In several cases the limit is set on the basis of density by visual observation. Whatever limit is considered fair, it means that actual sampling is necessary to determine compliance or violation. These are relatively lengthy procedures and the number of tests that can be made is limited.

GASES

In a number of cities the ordinances regulate the emission into the atmosphere of noxious and obnoxious gases. The only product of the combustion of fuel that is concerned is sulfur dioxide. Most of the sulfur in stack gases is in this form. Such a regulation opens up a wider field, however,—industrial gases, which locally may in some cases reach a concentration that is an annoyance, or even a health hazard. There are methods of determining the concentrations of the various gases that may be encountered.

ANTI-SMOKE ORDINANCE

Consideration should now be given to the so-called *anti-smoke ordinance*, which is aimed at control of products of combustion. Whether or not more than dense smoke is specified, the enforcement officer finds that his work includes abatement of fly-ash and sulfur gases. And whether or not private dwellings are exempt, an appreciable percentage of his time and that of his inspectors will be devoted to domestic smokers. Therefore, it is logical and proper that, in addition to the regulation of smoke, excessive fly-ash and sulfur gases should be prohibited. As stated previously, this prohibition is found in the later ordinances in effect in many cities. There should be no exemption of any class of fuel-burning plants.

Permissible Limits. As the limitation of dense smoke to 6 min during cleaning or building a new fire, with none allowed at any other time, is considered satisfactory in a high percentage of large industrial cities, such a provision would seem to be in line with current good practice. In addition, smoke of medium density (No. 2, Ringelmann Chart) could be limited to 5 or 6 min in an hour.

Excessive fly-ash should be prohibited, but to agree on what the limit should be in fairness to all concerned is difficult. The most recent suggestion is 0.5 grains per cubic foot of gas on stoker-fired installations and 0.75 grains on pulverized coal-fired installations. There must also be a simple method of checking in order that this part of the ordinance may be properly enforced.

The situation with regard to sulfur gases is different. There is at present no practicable method, for use in medium size and smaller plants, of neutralizing or eliminating these gases. One way to reduce the total amount of sulfur that gets into the atmosphere is to place a limit on the percentage of sulfur in fuel that may be burned, which is now being done in St. Louis.

ENFORCEMENT

Enforcement is the greatest barrier to real success in smoke abatement. An ordinance can be specific as to what is permissible and what is prohibited, but without adequate means of enforcement it is worth comparatively little. The number of inspectors in the large industrial cities varies from 1 to 60,000 population to 1 to 300,000 or more.

An inspector can cover only a limited area in a day. Regularly he has inspections to make, violations to investigate, complaints to follow up, firemen to instruct; besides other duties that come up occasionally. Corps are too small to keep up efficiently with their special duties, aside from smoke observations on industrial and commercial plants; this is exclusive of private homes, which introduce special problems. Conditions cannot only be interpreted on the basis of the reports of the smoke inspectors.

City Councils have allocated to smoke abatement such proportion of total income as they consider fair. A request for a sufficient increase in personnel to make possible the kind of supervision which the abatement officer would like to have would be met with refusal as being out of proportion with other necessary services.

Since it is not possible to do a complete job by the methods that have been in use, obviously new or different ways must be developed. Although considerable thought has been given to the matter, the extent of this paper does not permit discussing it in any detail. Such a plan naturally would regulate really to *prevent* violations, and would involve more specific regulations and place more responsibility upon owners.

To meet the demands of the public, it is going to be necessary, first, to draft laws that will make possible the measure of air purity it desires; secondly, to bring people to a realization of the fact that the low pressure heating plant as used in small commercial establishments, apartments and private houses is, in proportion, the worst offender; and thirdly, to secure their cooperation—willingly in most cases, arbitrarily in some.

INDUSTRIAL DUSTS, FUMES AND GASES

Another phase of air pollution which has been receiving attention is that which comes from raw materials used in industry and from industrial processes. Its control within the plant is an engineering job in air conditioning. That part of it will not be discussed in this paper, but where dusts, fumes and gases are in such concentrations as to be noxious or obnoxious they should not be exhausted into the atmosphere, but should be disposed of in some other manner.

CONCLUSION

The purpose of this paper will have been accomplished if the writer has convinced the reader of his own belief that the abatement of nuisance-making

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and health-damaging air pollution will go ahead at a greatly accelerated pace. If the conditions and their causes are analyzed, which can be done from training and experience, it is felt that readers will put their shoulders to this wheel of progress.

It is a problem in public relations, with many groups involved. There will be new and more stringent regulations, but there is no reason to anticipate that these will be in any way unfair. The rest is a matter of cooperation from interested groups—manufacturers of equipment, producers of fuel, engineers, and finally the users, who make up the public.

No. 1121

SMOKE-PRODUCING TENDENCIES IN COALS OF VARIOUS RANKS

By H. J. Rose * (*MEMBER*), Pittsburgh, Pa. and F. P. Lasseter,** Westport, Conn.

THE United States contains about half of the world's known coal supply, located in more than 30 states. Therefore coals of more than one variety are commercially available in most localities. Standard specifications are now available [1] for classifying coals according to rank, that is, according to their degree of progressive alteration in the natural series from lignite to anthracite. The increasing attention given to the health and economic aspects of air pollution by smoke justifies consideration of the smoke-producing potentialities of coals of various ranks.

Anthracites, because of their chemical composition, are inherently smokeless under all conditions of use. Smoke production from fuels of lower rank will depend not only upon the characteristics of each fuel but also upon the conditions under which it is burned. Under ideal conditions, any fuel can be burned without smoke, but in actual practice many fuels present a major smoke problem to designers and users of heating equipment.

It has long been recognized that the source of coal smoke lies in the tar vapors and hydrocarbon gases in the volatile matter produced when coal is heated. These volatile products may be liberated so rapidly in the furnace, and occupy such a large volume, that it is difficult to supply and mix in the necessary air, and to provide enough space for complete combustion. The result is smoke, a mixture of acrid, brownish tar vapors with black soot produced by the incomplete combustion of tar and coal gas.

The magnitude of the problem can be realized from the fact that typical high-volatile bituminous coals when heated will quickly yield a mixture of tar vapors and coal gas representing one-third to one-half of the total heating value of the original coal. This is well illustrated by the top sections of Figs. 1

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* Senior Industrial Fellow, Anthracite Fellowship, Mellon Institute of Industrial Research.
** Formerly Industrial Fellow, Mellon Institute of Industrial Research.

[1] Standard Specifications for Classification of Coals by Rank. *ASTM* Designation: D388-38. (1938 Supplement to Book of *ASTM* Standards.)

Presented at the 45th Annual Meeting of the American Society of Heating and Ventilating Engineers, Pittsburgh, Pa., January, 1939.

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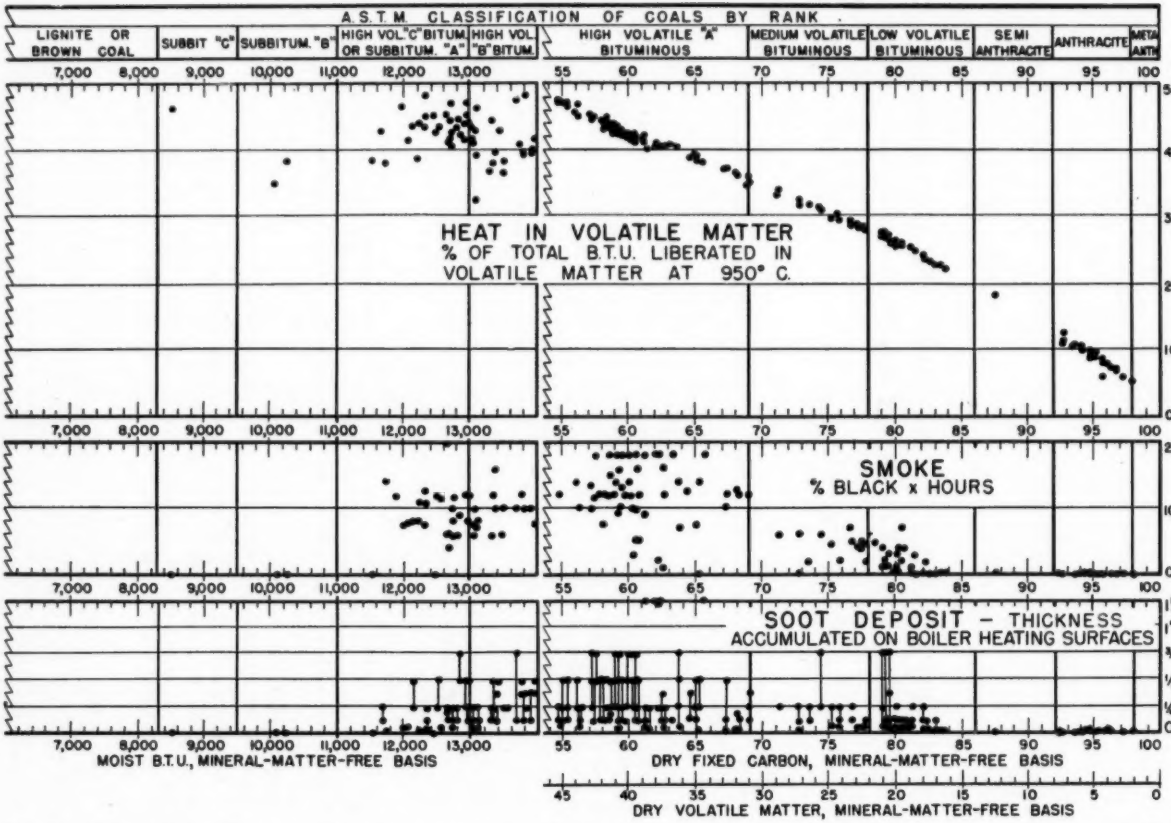


FIG. 1. CORRELATION OF RANK OF COALS WITH PER CENT OF TOTAL BTU LIBERATED IN VOLATILE MATTER, AND SMOKE AND SOOT PRODUCED IN HOUSE-HEATING BOILER TESTS. (BASED ON U. S. BUREAU OF MINES BULLETIN 276)

and 2, which show the percentage[a] of the total calorific value which appears in the *volatile matter*,[2] when coals of various ranks are heated according to the standard test.

For the higher ranks of coal there is a definite linear relationship between this factor and rank, which may be summarized as follows:

RANK OF COAL	PER CENT OF TOTAL BTU LIBERATED IN VOLATILE MATTER AT 950C
Anthracite	5–14 per cent
Semi-anthracite	14–21 per cent
Low-volatile bituminous coal	21–28 per cent
Medium-volatile bituminous coal	28–36 per cent
High-volatile bituminous coal *A*	36–47 per cent

The lower-rank coals do not show such a definite relationship, and the percentages range from about 48 per cent down to 30 per cent, with a tendency to decrease towards the lowest ranks.

While the relationship just referred to is interesting, it does not directly measure smoke-producing tendencies. All of the coals referred to yield some volatile combustible matter, yet some of them were known to be smokeless under all conditions of use. Therefore, in order to compare these data with practical smoke-producing tendencies, it was desired to locate information on the burning characteristics of coals of known analysis from all parts of the United States. The most suitable source of information seemed to be a U. S. Bureau of Mines bulletin[3] which summarized more than 500 burning tests in house-heating boilers, of coals from 22 states (plus a few tests of coals from five foreign countries). For the present purpose, some of the original results have been recalculated and graphed in Fig. 1, in order to present *practical* data on smoke and soot. The values in the top section of this figure refer to the same coals as those used in the burning tests.

The tests described in the bulletin were made in four round hand-fired domestic boilers, of either cast-iron or steel construction, which were arranged for steam or hot-water heating. The nominal grate diameters were 23 and 26 in. About 40 per cent of their rated capacity was usually carried by the boilers, and the tests usually lasted for more than 36 hours. Coal was charged in weights such as to give a firing period of 6 to 12 hours. The spreading method was used for anthracite, and alternate side-coking and center-cone methods were used for bituminous coals.

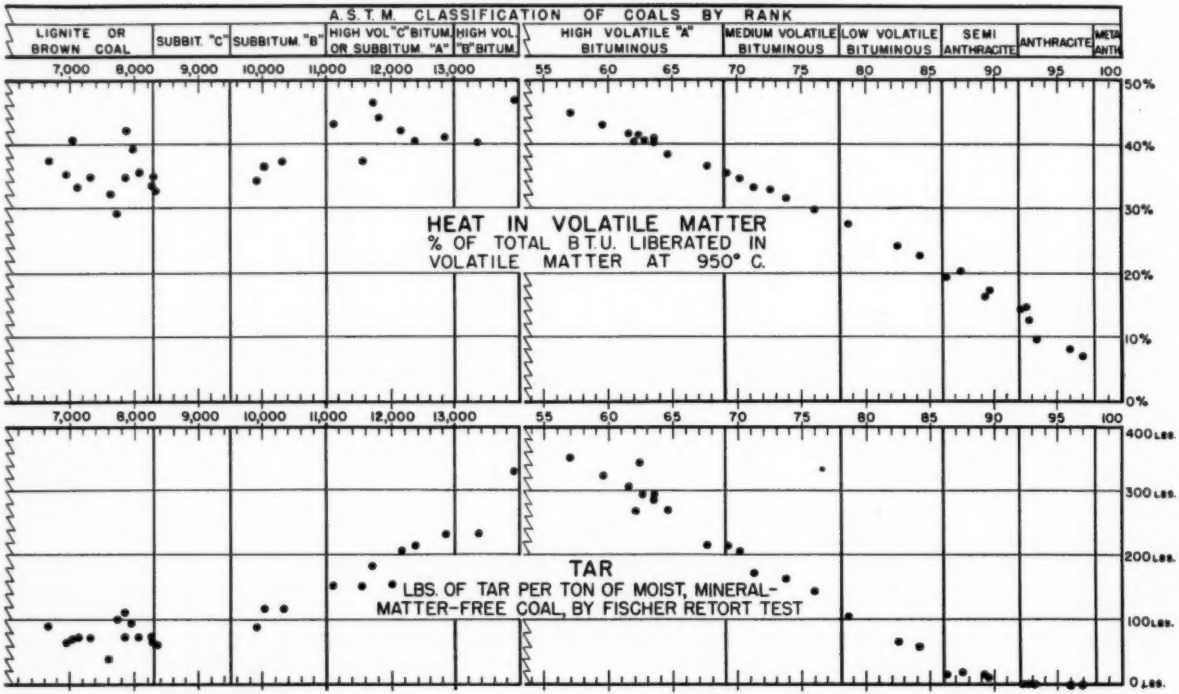
Smoke density was estimated from the Ringelmann-chart scale, and reported in the bulletin as average per cent black over a specified number of hours;

[a] These percentages are approximations calculated from the ordinary proximate analysis and Btu figures, as follows: An average heating value of 14,500 Btu per pound was assumed for the *fixed carbon* residue in the volatile-matter test. The yield of fixed carbon was therefore multiplied by 14,500 and the product was divided by the heating value of the original coal. This result, when subtracted from 100 per cent, obviously gave the approximate percentage of total heating value which was contained in the volatile matter.

[2] Standard Methods of Laboratory Sampling and Analysis of Coal and Coke. *ASTM* Designation: D271-33. (1936 Book of *ASTM* Standards.)

[3] Five Hundred Tests of Various Coals in House-Heating Boilers, by P. Nicholls, S. B. Flagg and C. E. Augustine. (*U. S. Bureau of Mines Bulletin* 276, 1928.)

FIG. 1. CORRELATION OF RANK OF COALS WITH PER CENT OF TOTAL BTU LIBERATED IN VOLATILE MATTER, AND SMOKE AND SOOT PRODUCED IN HOUSE-HEATING BOILER TESTS. (BASED ON U. S. BUREAU OF MINES BULLETIN 276)


A. S. T. M. CLASSIFICATION OF COALS BY RANK
LIGNITE OR BROWN COAL
SUBBIT. "C"
SUBBITUM. "B"
HIGH VOL. "C" BITUM. OR SUBBITUM. "A"
HIGH VOL. "B" BITUM.
HIGH VOLATILE "A" BITUMINOUS
MEDIUM VOLATILE BITUMINOUS
LOW VOLATILE BITUMINOUS
SEMI ANTHRACITE
ANTHRACITE
META ANTH.
7,000 8,000 9,000 10,000 11,000 12,000 13,000
55 60 65 70 75 80 85 90 95 100
50%
40%
30%
20%
10%
0%
HEAT IN VOLATILE MATTER
% OF TOTAL B.T.U. LIBERATED IN VOLATILE MATTER AT 950° C.
400 LBS.
300 LBS.
200 LBS.
100 LBS.
0 LBS.
TAR
LBS. OF TAR PER TON OF MOIST, MINERAL-MATTER-FREE COAL, BY FISCHER RETORT TEST

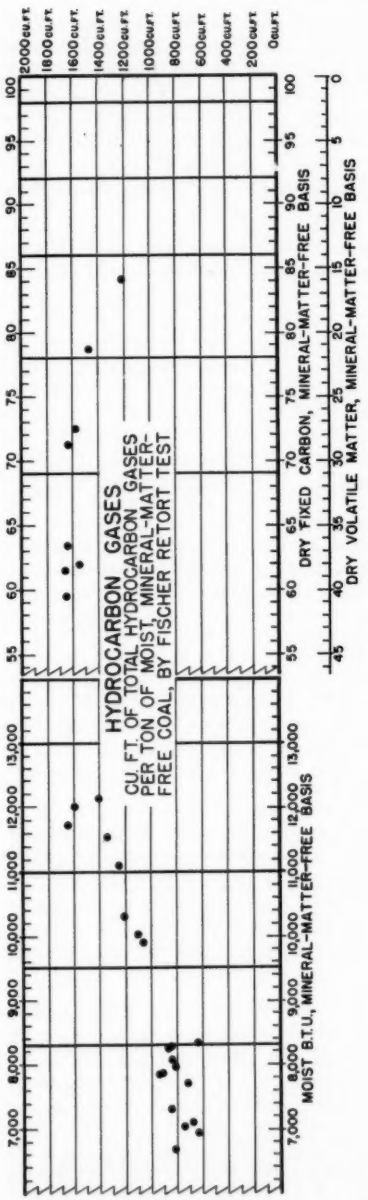


FIG. 2. CORRELATION OF RANK OF COALS WITH FISCHER RETORT YIELDS OF TAR AND HYDROCARBON GASES. (PER TON OF COAL ON THE MOIST, MINERAL-MATTER-FREE BASIS)

for example, "the smoke was 20 per cent black for an average of seven hours." The present writers have condensed this for the graph by multiplying the *average per cent black* by the number of hours, so that the previous example would be shown on the middle section of Fig. 1 as 140 (i.e. 7 x 20 per cent). The boiler heating surfaces were cleaned before each test was started, and the thickness of the soot accumulation was noted at its conclusion. This thickness, or range of thickness, is presented in the bottom section of Fig. 1, and is seen to have varied up to 1¼-in. thickness, for tests which never lasted more than 50 hours.

These graphs show that neither smoke nor soot were formed when burning anthracites or the one sample of semi-anthracite. Under the test conditions, both smoke and soot became apparent with many low-volatile bituminous coals beginning at about 82 per cent fixed carbon (18 per cent volatile matter) on the dry, mineral-matter-free basis, which is equivalent to about 16 or 17 per cent volatile matter on the usual *as fired* basis. As would be expected, high-volatile bituminous coals produced the most smoke and soot. In the zone comprising high volatile *C* bituminous and sub-bituminous *A* coals (which can only be differentiated by agglutinating or weathering properties), the smoke and soot diminished and little was noted from the three samples of sub-bituminous coals of still lower rank. No lignite samples were included in this series of tests.

The individual values for smoke and soot on Fig. 1 should not be accepted too literally for a number of reasons which need not be enumerated here. However, the graphs are believed to be of interest and value because they show general trends, irrespective of the fact that the size of coal and certain other important variables were not taken into consideration.

The most satisfactory method of determining the smoke-producing potentialities of a fuel is to burn it under the precise conditions which are of special interest. For example, Limbacher and Sherman [4] have obtained continuous records of smoke concentrations resulting from burning bituminous coals by use of small underfeed stokers. They used a scale of units which is directly proportional to the concentration of smoke.

Nevertheless there is always a desire to find some laboratory index of smoke-producing *tendencies* which can be applied to coal itself, irrespective of the conditions under which it may be burned. Various laboratory smoke tests have been proposed in which coal is heated in a current of air. They have the obvious defect that the amount of smoke formed depends upon the particular conditions selected, such as air supply, furnace temperature, condition of sample, etc.

Since smoke originates from the tar and hydrocarbon gases liberated from coal, it is logical to study how the yields of these products vary with rank. For this purpose it was desired to use tar and gas yields obtained at a rather low temperature before extensive cracking reactions had occurred. Therefore low-temperature carbonization yields, made in a Fischer retort, were selected.

[4] The Performance of Several Types of Bituminous Coal on Small Underfeed Stokers, by H. R. Limbacher and R. A. Sherman. (Mimeographed report published by *Bituminous Coal Research, Inc.*, Washington, D. C.)

In this test,[5] the coal sample is placed in a thick-walled aluminum retort and heated according to a definite schedule to the desired final temperature, which is usually 500 C (932 F). The yield of dry tar is obtained, and the gases may be analyzed.

Fig. 2 presents data for a series of coals ranging from anthracite to lignite (but different from those shown in Fig. 1). Some of the coals were tested by the writers; the other results being obtained from publications by U. S. Bureau of Mines authors.[6, 7, 8] The middle section of Fig. 2 shows that the anthracite produced no tar whatever, and the semi-anthracite only a small amount (down to a trace) of tar. This was to be expected; in fact Turner [9] has proposed a simple and sensitive tar test to differentiate between true anthracites and semi-anthracites in border-line cases.

It is also seen that the tar yield from low-, medium- and high-volatile bituminous coals increases regularly from about 50 to 350 lb of tar per ton of ash-free coal. The latter figure amounts to 17.5 per cent by weight, or about a barrel of tar per ton of coal, which forcibly illustrates the problem which confronts the user who tries to burn such coal smokelessly in small heating equipment. A single cubic foot of such coal may yield a gallon of tar and nearly 40 cu ft of hydrocarbon gases. The tar yield falls off steadily in the lower-rank bituminous and sub-bituminous groups, and averages less than a hundred pounds per ton for the lignites. However, it must not be forgotten that these low-rank coals have such a low heating value that a much larger tonnage (twice as much in the case of lignite) must be burned in order to obtain the same amount of heat as would be obtained with high-rank coals.

The correlation of tar yields with the graphs of smoke and soot production in Fig. 1 is as good as would be expected in view of the many variables affecting the latter values. The range in rank of coals yielding tar is somewhat greater than the range of coals yielding smoke and soot in the practical tests in hand-fired boilers. This is logical, since any furnace should be able to consume at least a small amount of tar vapors without appreciable smoke production.

The total volume of hydrocarbon gases per ton of coal produced in the Fischer retort test is shown in the bottom section of Fig. 2. This information was not available for all of the samples. However, it is known from published tests (made under other conditions) that the yield of hydrocarbon gases obtained from anthracite is small.[10] High-volatile bituminous coals produce as much as 1600 cu ft of such gases per ton of ash-free coal, thus adding to the problem of providing sufficient air and furnace volume for complete combustion.

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[5] Methods and Apparatus Used in Determining the Gas, Coke, and By-Product Making Properties of American Coals, by A. C. Fieldner, J. D. Davis, R. Thiessen, E. B. Kester and W. A. Selvig. (*U. S. Bureau of Mines Bulletin* 344, 1931.)

[6] Loc. Cit. See Note 5.

[7] *U. S. Bureau of Mines Technical Papers Nos.* 511, 519, 524, 525, 531, 542, 543, 548.

[8] Low-Temperature Carbonization of Lignites and Sub-Bituminous Coals, by J. D. Davis and A. E. Galloway. (*Industrial and Engineering Chemistry*, Vol. 20, 1928, p. 612.)

[9] Anthracites and Semianthracites of Pennsylvania, by H. G. Turner. (*Transactions Coal Division, AIME*, Vol. 108, 1934, p. 330.)

[10] Volatile Matter of Pennsylvania Anthracite, by H. G. Turner and W. L. Keene. (*Industrial and Engineering Chemistry*, Vol. 27, 1935, p. 1373.)

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Naturally there are variations in the compositions, as well as the yields, of tar and gas produced from such a wide range of coals, and these have some effect on the problem of smokeless combustion. However, the subject will not be developed here.

Others have reported that smoke production is proportional to the *volatile matter* of coal, which may well be the case with a limited range of coals, particularly of the bituminous class. However, the studies on which this paper is based have shown that the volatile matter percentage considered alone is not a suitable smoke index when coals of all ranks are included.

When it is desired to rate coals according to their smoke-producing tendencies, but without specifying particular combustion conditions, the tar yield in the Fischer retort test [11] should be a practical and sufficiently accurate index.

DISCUSSION

W. C. KELLY: I would like to ask Mr. Rose if there are any data available on the amount of tar in the various coals, if these data can be secured and if so, where?

H. J. ROSE: Considerable information on tar yields will be found in various recent publications of the U. S. Bureau of Mines, such as those mentioned in footnotes 5, 7 and 8 of this paper.

In recent years the Bureau of Mines has made a systematic study of the gas, coke and by-product making properties of coals from many different states and seams. As a part of that study they have obtained tar yields not only in the Fischer retort, mentioned in the paper, but they have also obtained the tar yields over a wide range of coking temperatures (from 500 to 1100 C) by another method. This is probably the most complete source of published information on tar yields.

Of course, the work which might be done is enormous, since there are about 6,000 operating bituminous coal mines in the United States. However, as far as they have gone, the work is being done very thoroughly, and will give splendid material for making comparisons provided that the particular coals that might interest you have been tested.

MR. KELLY: Would you signify which of the papers mentioned most thoroughly covers the field?

MR. ROSE: Monograph 5 of the U. S. Bureau of Mines gives data on about 30 coals. Most of the other papers refer to coal from a particular seam and mine, so that it is just a question as to what coal you are interested in. The character of the papers is otherwise very similar.

P. H. WEITZEL: I would like to know if there is a tendency to produce more smoke in a hot-water boiler with a cast-iron jacket surrounded by water, than in the furnace where there is a much higher fire-box wall temperature.

MR. ROSE: Undoubtedly the difference in conditions will result in some difference in smoke production. The problem is, of course, to keep the volatile products hot enough for ready combustion, to mix in sufficient air, and to provide combustion space. Difference in radiant energy and furnace temperatures in the two types of equipment may be sufficient to cause substantial differences in smoke production. Certainly the combustion of the large volume of tar vapor and hydrocarbon vapor

[11] Loc. Cit. See Note 5.

which is present with certain coals is a problem that tries the skill of the designer and the user. Most any difference in furnace condition such as temperature of walls or turbulence of the gases would have an effect on smoke.

It may be of interest to learn some personal reactions as to just what stokers mean in domestic heating. Several years ago I had occasion to buy a home here in Pittsburgh, and when I bought the home it was heated with gas. Automatic gas heat is, of course, a very nice fuel. We will not deny that; especially if you can overlook the cost factor. But, being in the solid fuel industry, I was not satisfied to use that indefinitely. Not long ago I had occasion to install what we call a completely automatic anthracite stoker.

That installation consists of a 10-ton dustless coal bin built of 2 × 4's and fir plywood in the corner of the basement. The coal screw extracts rice anthracite, taking it to the furnace, and burns it there in an under-feed retort where the ash spills over the edge and drops into a pit. That is a point that I find a few people do not understand. Instead of having an ashcan under there, we built under by installation a large storage pit. I have not seen, and no one in my family has seen, any ashes around my place for weeks, and they are not expected to be seen for at least two or three months. The bin is large enough to hold all the ashes for a third of a year.

The whole of the equipment, of course, is thermostatically controlled, set at 69 F, and, so far as I am concerned, I have never looked at the thermostat at any time when it was more than ±1 deg. The installation was put into the home in midwinter, without any appreciable interruption in the heat supply. The pit was dug under an existing boiler, an old cast-iron boiler that is 35 years old, without disturbing, moving, or breaking any connections. The people upstairs in the house did not notice the interruption in the heat. The old fire was pulled and the new stoker was put in and it is operating now in a way that has been very interesting to me because we had that immediate comparison with automatic gas. I waited for some days to get the reaction from my family, but outside of the satisfaction with the heat, I received no comment whatever. I began to question whether any more dirt, coal dust or ash was noticeable, but the response was in the negative. I have been noticing the last couple of weeks that the family wash is now being hung in the basement within 24 in. of the coal bin and of the stoker, for my family claims it is much cleaner in that basement than it is outdoors in Pittsburgh.

Pittsburgh is not what you might call a natural anthracite territory, so the question of cost may arise. I find that the cost of rice anthracite delivered in my home is very close to the cost of the domestic sizes of hand-fired coke; that is, stove or chestnut coke delivered in the basement here in Pittsburgh. It is a surprise to everybody when they learn that. The cost is going to be considerably less than that of the cheap natural gas in Pittsburgh.

I have burned fuels from many states under a great many conditions previously, mostly hand-fired, and to have a completely automatic coal heating plant that requires no attention whatever for months at a time is a satisfactory experience.

What am I going to do with this ash when it accumulates? About three times a year it will be dampened and removed, but it will not be carried away from the premises. For more than a year we have been carrying on an extensive investigation of the uses of anthracite ash for agricultural purposes, for loosening up the soil and improving the texture and moisture balance. It has been used for that purpose for many years, but nobody collected any data, and we decided we could get some facts. We have been rather surprised at the results we have obtained. They are so interesting that my wife is not satisfied with the rate we are producing anthracite ash in the basement, and she has made a dicker with the

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nextdoor neighbor to get her's, because it is going to be a number of years before the clay soil in our 50-ft city lot will have enough ash. We put in bales and bales of German peat, not with entirely satisfactory results, but now we are going to use ash, and I think we are going to get some gratifying results.

E. T. Selig, Jr., who is a member of the Society, and whom many of you know, installed the first heating plant of this sort in his home in Harrisburg. When he came with us at the Mellon Institute, Pittsburgh, he rented that home complete, with heat included, winter heat and complete year-round hot water service from this stoker, which operates 12 months a year. This added a small monthly charge to the rent. He made an arrangement with the coal dealer to keep the coal bin filled; the stoker service man makes periodic calls; he tells the ash collector when to come and haul away the ash. The tenant of that house has paid no attention to the heating system, in fact, he has been instructed to let it alone; it is all taken care of for him, and for a very modest monthly sum.

After it had been in operation for a considerable period of time, Mr. Selig was in Harrisburg and went to visit the tenant and he asked him what he thought of the heating plant. The man looked rather blank and told him he had practically forgotten about it. That shows what can be done with solid fuel.

This morning one of the discussers deplored the lack of progress that has been made in solid fuel heating equipment, and, to a certain extent, his remarks were true and well put, but I do want to say that the day has already arrived when we have an entirely new standard of convenience and cleanliness in the use of solid fuels in the home. I think that it is a very fortunate thing that the Society has recently authorized the formation of a Committee on Solid Fuels to study, improve, and bring to the attention of this Society, and the public, the many interesting things that are being developed in the use of solid fuel.

No. 1122

AIR FILTER PERFORMANCE AS AFFECTED BY LOW RATE OF DUST FEED, VARIOUS TYPES OF CARBON, AND DUST PARTICLE SIZE AND DENSITY

By FRANK B. ROWLEY * AND RICHARD C. JORDAN ** (*MEMBERS*)
MINNEAPOLIS, MINN.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the Engineering Experiment Station, University of Minnesota

THIS paper presents the results of a research program which is a continuation of the work described in a previous paper [1] presented at the Semi-Annual Meeting of the Society, June, 1938.

The object of this part of the program was to determine the dust arrestance for low rates of dust feed through the filter, and filter performance when using different types of carbons, dusts of different particle sizes, and dusts of different densities. The test apparatus and procedure were described in the previous paper. The apparatus, with the exception of the dust feeding device, was the same as that described in the ASHVE Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work. The photograph, Fig. 1, shows a general view of the test equipment as set up and used throughout the tests.

As in the previous test program four filters were selected as typical of those used in practice, and were used throughout all of the tests. These filters have been designated by the letters *A*, *B*-1, *C*, and *D* and were described in detail in the previous paper. Briefly they may be described as follows:

A—A permanent type of cleanable oil filter, 4 in. thick, with 24 layers of expanded metal and wire screen graded from coarse mesh at entrance to fine mesh at leaving side.

B-1—A viscous coated throw-away type filter 2 in. thick. The fibrous media were graded in fiber size, density, and oiling from entering to leaving side. This filter is

* Director. Engineering Experiment Station, University of Minnesota.
** Instructor Engineering Experiment Station, University of Minnesota.

[1] Air Filter Performance as Affected by Kind of Dust, Rate of Dust Feed and Air Velocity Through Filter, by Frank B. Rowley and Richard C. Jordan. (ASHVE TRANSACTIONS, Vol. 44, 1938, p. 415.)

Presented at the 45th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Pittsburgh, Pa., January, 1939.

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designated in this report as *B*-1 due to the fact that slight changes had been made in the oiling and fiber diameter from those filters used in the previous report. These changes, however, did not apepar to make any appreciable difference in the test results.

C—A cellular type filter 2 in. thick. The filter was built in two sections each with the axis of cells set at 45 deg to the center line of duct and at 90 deg to each other. The cells on the entering side were of larger dimension than those on the leaving side of the filter.

D—A filter of cotton media of coarse material on the entering side and glazed on the leaving side. The filter media were accordion plaited in frame to give an area of approximately 12 times the cross-sectional area of air stream.

For convenience the results given in this paper have been grouped under six headings as follows:

1. Rate of Dust Feed and Filter Arrestance.
2. Dust Particle Size and Filter Performance.
3. Different Types of Carbon and Filter Performance.
4. Dust Density and Filter Performance.
5. General Comparisons of Filter Characteristics.
6. Conclusions and Recommendations.

RATE OF DUST FEED AND FILTER ARRESTANCE

The standard test code for rating air filters calls for 0.35 g (grams) of dust per 1000 cu ft of air. This corresponds to a rate of 17.7 g per hour when using a 20-in. square filter with a face air velocity of 300 fpm. In the first series of tests the rate of dust feed was raised to 20 g per hour for a 20-in. filter with 300 ft face velocity to provide a practical rate of feed to meet the code requirements. When the test results for this rate of dust feed were compared to those for rates of 40 and 60 g per hour, other conditions remaining the same, it appeared that much higher rates of feed might be justified in the standard code.

The lowest rate of dust feed used in the first series of tests was 10 g per hour, and the question arose as to whether the filter performance at high rates of dust feed, selected for test purposes, would be comparable to the performance characteristics which might be expected on the filter at the very low rates of feed met in practice. In order to answer this question a series of tests was run in which all filters were tested at a dust feed of 2 g per hour, other conditions remaining constant and equal to former tests. The dust mixture for this series consisted of 50 per cent Pocahontas ash screened through 200-mesh screen, 20 per cent Illinois fly-ash passing a 200-mesh screen, 20 per cent Germantown Eagle Brand lampblack passing a 100-mesh screen, and 10 per cent Fuller's Earth passing a 100-mesh screen. The results of this series of tests together with the previous tests ranging from 10 to 100 g per hour of the same dust mixture are shown in Table 1. The corrected test values shown under the 2 g rate take into consideration the weight of dust in the laboratory air. To obtain the additional weight the test apparatus was run without a filter in place and with no artificial dust feed. The increase in crucible weight was recorded for four 5-hour periods. The recorded increases were 0.0015, 0.0013, 0.0017, and 0.0015, giving an average of 0.0015 g for the 5-hour period. As a check on this method, a filter, which from previous tests was estimated to have an arrestance of 80 per cent, was placed in the test apparatus and a test run was made also without the addition

of the dust mixture to the air stream. The gain in crucible weight was found to be 20 per cent of that without the filter, indicating that the calibration tests were correct, and that the weight of dust in the laboratory air used for the filter tests increased the dust fed to the filter by approximately 0.248 g per hour. This correction was made for the 2-g rate, but not for the higher rates shown in Table 1. This correction obviously would have much less effect for the higher rates of dust feed.

Later in this test series it was discovered that the lampblack used was of slightly different density and other characteristics than that used in the first

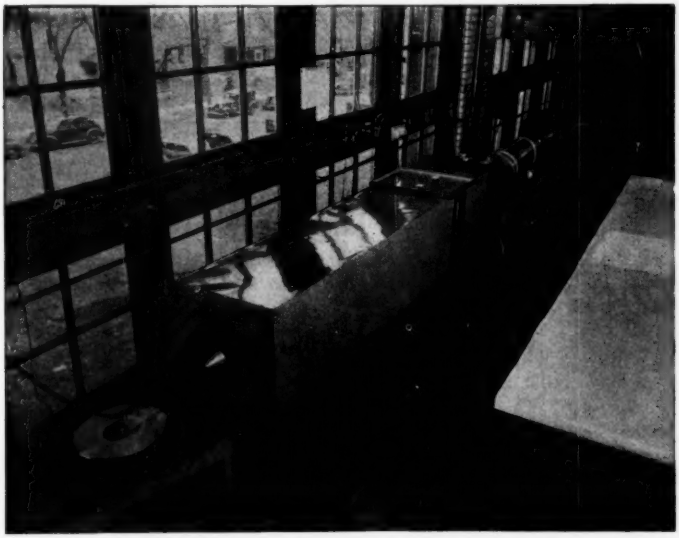

FIG. 1. ASSEMBLED VIEW OF TEST APPARATUS

series, and that the dust arrestances for Filters *A*, *B*-1, and *C* were materially higher for this lampblack when used alone than for the same filters on the previous lampblack. Since only 20 per cent of lampblack was used in the dust mixture of the present tests, this difference on the final results would not be great, but it would have a tendency to raise the arrestance of those tests made at the 2-g rate with the new lampblack. It should be noted that the rate of 2 g per hour as fed corresponds to 0.617 grains per 1000 cu ft, which is comparable to the dust concentration in the air of industrial districts. This rate of feed is only eight times the normal dust concentration in the laboratory air, whereas a feed of 40 g per hour is 160 times normal.

The conclusion drawn from the tests shown in Table 1 is that for typical filters the arrestance values at low rates of dust feed are substantially the same as those for the 10- to 100-g rates. Since the higher rates are more practical for actual filter performance tests, it is reasonable to use them. The

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TABLE 1—AVERAGE DUST ARRESTANCES FOR FILTERS *A*, *B*, *C*, AND *D*
(*Face air velocity* 300 *fpm*. *Dust mixture* 50, 20, 20, 10)

FILTER	RATE OF DUST FEED, GRAMS PER HOUR						
	2	10	20	30	40	60	100
A	77.5 (Corrected 80.1)	..	77.2	..	77.5 80.3	76.9	79.7
B	82.1 (Corrected 84.3)	80.0	78.2	81.3	81.0 81.9 81.7	82.0	82.8
C	71.6 (Corrected 75.2)	72.3	72.0	..	69.9	71.4	68.2
D	88.4 (Corrected 90.7)	89.8	92.5 92.0	..	93.5	94.6	92.4

tests at the 2-g rate were not continued for a sufficient length of time to determine the effect of this low rate of dust feed on the life of filters, but obviously such tests are virtually impossible from the standpoint of time required.

DUST PARTICLE SIZE AND FILTER PERFORMANCE

The object of this part of the program was to determine the particle size distribution for different kinds of dusts screened to different sizes, and to find the effect of particle size on the performance of different types of filters. The analysis for particle size was made for all dusts used excepting Fuller's Earth, but the effect of particle size on filter performance was only determined for Illinois fly-ash and Pocahontas ash for two different ranges of screenings.

TABLE 2—PARTICLE SIZE DISTRIBUTION IN DIFFERENT KINDS OF DUSTS

TYPE OF DUST		APPROX. AVG. DIAM. OF PARTICLE	APPROX. DIAM. OF PARTICLE WITH AVG. VOLUME	DISTRIBUTION			PER CENT VOL. WITH DIAM. MORE THAN 25 MICRONS
Material	Screen Mesh			Per Cent Less Than 5 Microns	Per Cent Less Than 15 Microns	Per Cent Less Than 25 Microns	
Carbon Black....	100	3.7	7.0	84.0	98.8	100.0	0
Lampblack.....	100	4.6	7.2	71.9	96.8	100.0	0
Bone Black.....	100	3.7	5.9	80.3	98.9	100.0	0
Cottrell Ash.....	200	2.8	6.0	96.7	99.6	100.0	0
Illinois Fly-Ash...	200 to 325	5.4	20.1	79.6	93.5	96.0	95.3
Illinois Fly-Ash...	325	4.4	9.7	80.9	94.9	98.3	57.5
Pocahontas Ash..	200 to 325	8.9	22.3	60.4	84.4	91.9	89.0
Pocahontas Ash..	325	3.9	10.3	84.6	96.9	98.9	43.8

The results of the dust particle size analysis are shown in Table 2. When one figure is given in the column headed *Screen Mesh,* it indicates that all

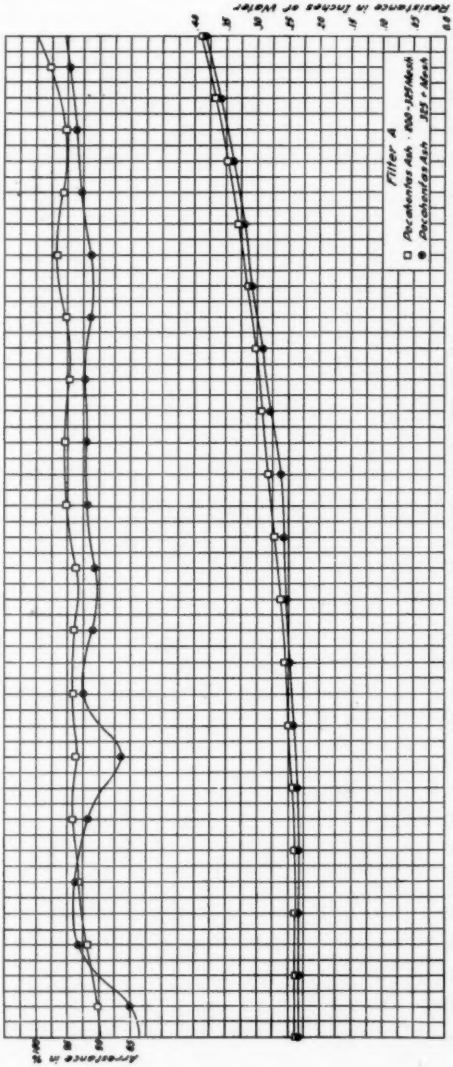
Resistance in Inches of Water
Filter A
Pocahontas Ash · 200-325 Mesh
Pocahontas Ash · 325 + Mesh
Height of Dust Feed in Grams
Arrestance in %

FIG. 2. EFFECT OF POCAHONTAS DUST PARTICLE SIZE ON PERFORMANCE OF FILTER *A*.
DUST FEED, 40 G PER HOUR

dust particles passed a screen of that mesh. When two figures are given it indicates that all dust particles passed the larger screen and were retained on

TABLE 3—COMPARISON OF FILTER PERFORMANCE ON DIFFERENT DUST PARTICLE SIZES[a]
(*Dust Feed* 40 *g per hour. Face air velocity* 300 *fpm*)

TYPE OF DUST	APPROX. DIAM. OF PARTICLE WITH AVERAGE VOLUME	FILTER *A*			FILTER *B*			FILTER *C*			FILTER *D*		
		Arres. Per Cent	Life Hours	D.H.C.[b] Grams	Arres. Per Cent	Life Hours	D.H.C.[b] Grams	Arres. Per Cent	Life Hours	D.H.C.[b] Grams	Arres. Per Cent	Life Hours	D.H.C.[b] Grams
Pocahontas Ash...	10.3	91.4	16.0	585.6	93.2	6.7	249.8	77.7	10.3	325.5	94.5	13.5	511.0
Pocahontas Ash...	22.3	94.4	16.0	604.1	93.5	7.4	277.4	84.9	12.9	441.8	91.8	12.7	464.0
Illinois Fly-Ash....	9.7	..	..	..	96.3	10.8	415.9	81.6	13.9	457.7	94.6	30.4	1152.8
Illinois Fly-Ash....	20.1	..	..	..	96.2	15.3	588.9	83.3	15.0	497.0	90.6	..	..

[a] Results based on a resistance rise of 0.15 in. of water across the filter.
[b] Dust holding capacity.

TABLE 4—COMPARISON OF FILTER PERFORMANCE[a] ON DIFFERENT CARBONS
(*Dust Feed* 40 *g per hour. Face air velocity* 300 *fpm*)

TYPE OF DUST	DUST DENSITY JOLTED		FILTER *A*			FILTER *B*-1			FILTER *C*			FILTER *D*		
	Grams/ cc	Pounds/ cu ft	Arr. Per Cent	Life Hours	D.H.C. Grams	Arr. Per Cent	Life Hours	D.H.C. Grams	Arr. Per Cent	Life Hours	D.H.C. Grams	Arr. Per Cent	Life Hours	D.H.C. Grams
Lampblack.....	0.203	12.7	75.9	4.8	146.6	66.0	1.8	45.2	45.4	2.8	50.8	82.0	0.9	29.5
Carbon black....	0.210	13.2	71.4	5.1	145.8	58.2	3.1	71.0	29.4	6.2	72.4	80.9	1.5	48.5
Bone black.....	0.741	46.3	78.7	26.9	838.4	81.8	11.8	386.0	58.2	12.5	292.4	84.7	10.8	365.4
Soot.....	0.135	8.45	..	..	..	58.9	..	..	29.7	..	..	66.9	..	..

[a] All values based on a resistance rise of 0.15 in. of water across filter.

the finer screen. In making the analysis of particle size from the screened samples a small portion of the dust was placed on a glass slide and covered by a drop of turpentine. The slide was then tapped until the dust particles had distributed themselves in the turpentine, after which the turpentine was evaporated leaving the dust particles exposed on the glass surface. Microphotographs were then made of these samples and used for measuring dust particle size. Results given in the table are the averages of from 300 to 500 particles for each sample of dust. Since the arrestance of a filter is based on the percentage, by weight, of the dust removed, it would appear that the figures in the column headed *Approximate Diameter of Particle with*

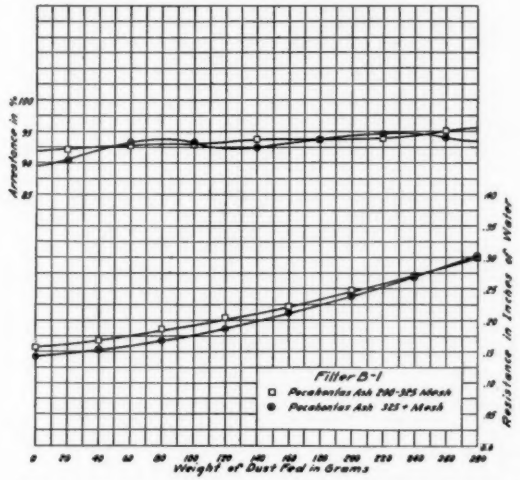
Arrestance in %
Resistance in Inches of Water
Weight of Dust Fed in Grams
Filter B-1
Pocahontas Ash 200-325 Mesh
Pocahontas Ash 325+ Mesh

FIG. 3. EFFECT OF POCAHONTAS DUST PARTICLE SIZE ON PERFORMANCE OF FILTER *B*-1. DUST FEED, 40 G PER HOUR

Average Volumes are better measures of the qualities of dust as a filter test material than are the figures giving the approximate average diameter of particles. The volume increases as the cube of the diameter increases and weights of the dust particles are in direct relation to their volumes.

Referring to the distribution of particle sizes in the various samples, it is interesting to note that for the carbon black, lampblack, bone black, and Cottrell ash, all of the particles are below 25 μ (microns) in diameter, even though the 100-mesh screen used for the first three samples would pass 147-μ diameter particles and the 200-mesh screen used for the Cottrell ash would pass 74-μ diameter particles. It should be noted, however, that carbon black and lampblack have a tendency to agglomerate and it is very difficult to pass them through a screen finer than 100-mesh, even though the individual particles should all pass a 325-mesh screen. The Cottrell dust could all have been screened through a 325-mesh screen as it does not agglomerate. In the case of the Illinois fly-ash and Pocahontas ash two different ranges of particle

sizes were screened. The figures giving particle size distribution show that a large number of the fine particles were retained on the 325-mesh screen. As a matter of fact the greater number of particles of Illinois fly-ash and Pocahontas ash which were retained on the 325-mesh screen should have

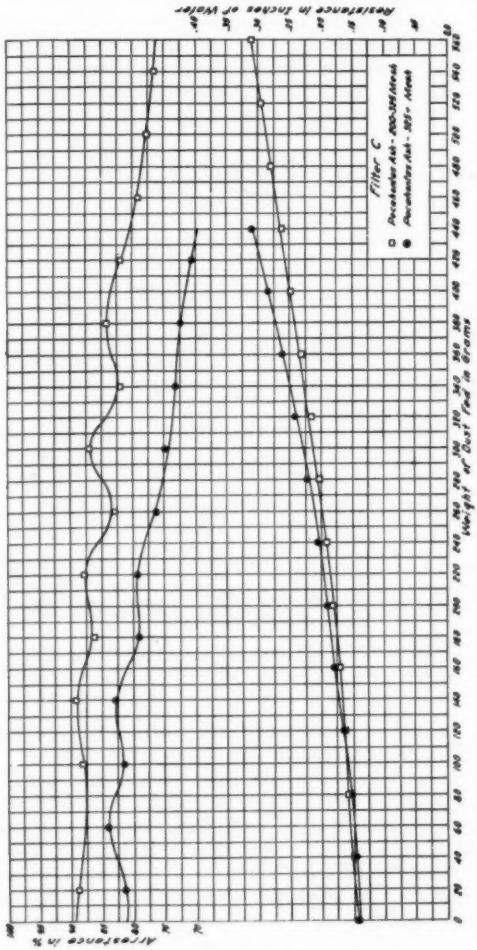
Filter C
Pocahontas Ash - 200-325 Mesh
Pocahontas Ash - 325+ Mesh
Resistance in Inches of Water
Weight of Dust Fed in Grams
Arrestance in %

FIG. 4. EFFECT OF POCAHONTAS DUST PARTICLE SIZE ON PERFORMANCE OF FILTER *C*. DUST FEED, 40 G PER HOUR

passed this screen. This retention was caused probably by many of the small particles adhering to larger ones and thus being prevented from passing the screen.

An interesting point to consider is the large percentage in total weight of dust samples accounted for by the very small percentage of particles over

25 μ in diameter. This is shown very clearly for the Illinois fly-ash screened between 200- and 325-mesh, and to a slightly lesser extent for the Pocahontas ash screened to the same size. This is a very important point to consider when selecting a dust to be used for testing a filter on the weight basis. A small percentage of large sized particles may have an undue influence on the filter arrestance. While a large percentage of the weight of the coarsely screened coal ash dusts are made up of particles above 25 μ in diameter, all of the particles of these samples were of the type which float readily in the air.

An attempt was made to use the dust from Illinois fly-ash and Pocahontas ash screened between 100- and 200-mesh, but many of these particles were so heavy that they settled out of the air very rapidly and were impractical for filter tests. The particle sizes which should pass the different screens are:

Screen Size	Dust Particle Size Microns
100	147
150	104
200	74
325	43
400	38

From the results of Table 2 it would appear that carbon black, lampblack and bone black might be screened through a 100-mesh screen, Cottrell ash either through a 200- or 325-mesh screen, and Illinois fly-ash and Pocahontas ash should be screened through a 325-mesh screen. From the table of screen sizes it is evident that nothing would be gained by screening finer than 325-mesh.

An attempt was made to grade further the particle of Illinois fly-ash and Pocahontas ash which passed through the 325-mesh screen by a process of elutriation. It was found that when these dusts were mixed with a solution of water and alcohol only about 1.2 per cent would remain in suspension for a period of one hour. As this was the minimum time that would show a definite difference in diameters of particles which settled out and which remained in suspension, the method was abandoned as impractical.

Table 3 shows the performance characteristics of each of the four filters when using the fine and coarse screened Pocahontas and Illinois fly-ashes. The test results for the Pocahontas ash are shown in the curves of Figs. 2, 3, 4 and 5 for Filters *A*, *B*-1, *C*, and *D* respectively. Illinois fly-ash was not used for Filter *A*. This was due to the time required and to the probability that nothing new would be gained over results for the other tests.

From the curves it will be noted that the dust arrestance was maintained uniformly throughout the test period for all filters excepting Filter *C*, in which case it gradually decreased as the test progressed. This also had a tendency to increase the life of the filter. The tests presented later on for the same filter with dust mixtures showed substantially uniform results throughout the test period at the same rate of dust feed. The reduction in arrestance cannot be interpreted, therefore, to indicate that 40 g per hour is loading the filters too rapidly when mixtures containing some carbon are used.

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An analysis of the test data in Table 3 and the curves indicates that the performance characteristics for Filters *A*, *B*-1, and *C* are slightly better when using the coarser ash, the greatest difference in arrestance being for

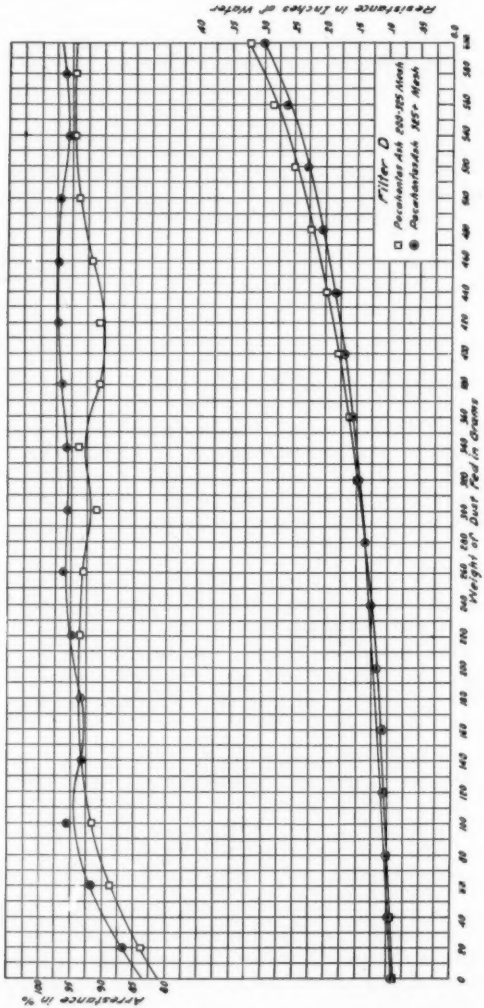
Filter D
Pocahontas Ash 200-325 Mesh
Pocahontas Ash 325+ Mesh
Arrestance in %
Resistance in Inches of Water
Weight of Dust Fed in Grams

FIG. 5. EFFECT OF POCAHONTAS DUST PARTICLE SIZE ON PERFORMANCE OF FILTER *D*. DUST FEED, 40 G PER HOUR

Filter *C*. For Filter *D*, however, the results are reversed. When using the Illinois fly-ash in Filter *D* it was found that the coarse ash had a tendency to drop off from the face of the filter making it impossible to complete the

test without giving misleading results. This was not true of the fine mesh dust as the cotton filtering pad was weighed before and after the test accounting for all but 4.95 per cent of the weight of dust taken out of the air by the filter. A part of this loss was accounted for by the dust which actually fell

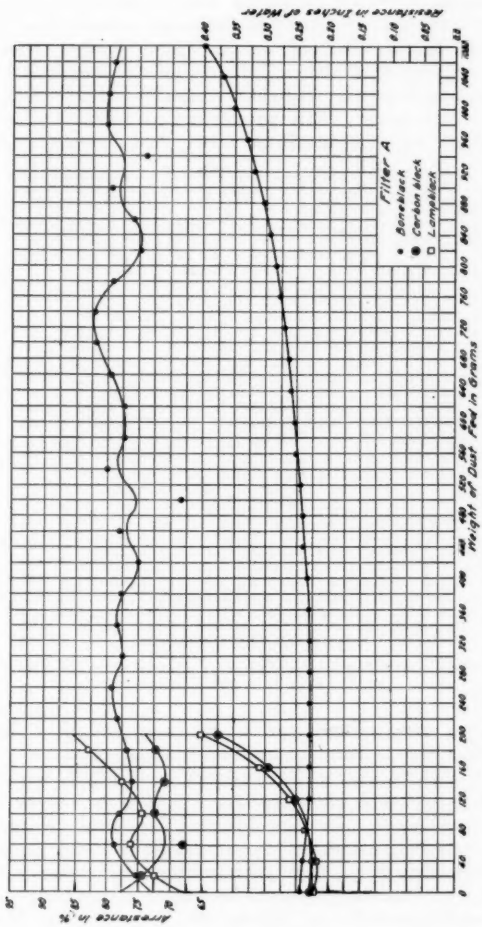

FIG. 6. EFFECT OF DIFFERENT CARBONS ON PERFORMANCE OF FILTER *A*. DUST FEED, 40 G PER HOUR

off from the filter media in removing it from the frame. The results of this part of the investigation, indicate that the finer dusts are better adapted to the rating of filters, and it is recommended that if either Pocahontas or Illinois fly-ash is to be used it should be screened through a 325-mesh screen.

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DIFFERENT TYPES OF CARBON AND FILTER PERFORMANCE

The filter test code states that the standard dust shall contain 50 per cent by weight of powdered lampblack containing 97.5 per cent of free carbon minimum, and having a bulking value of 3½ lb per cubic foot minimum. Previous investigations have shown that there are many different types of carbon that will fill this requirement and that some of them have a marked effect on the performance of a filter. Even though the specification is limited to lampblack there is still a wide latitude in materials which could be selected to meet the code requirement. This uncertainty as to the characteristics of a suitable lampblack and the handicap which large percentages of some

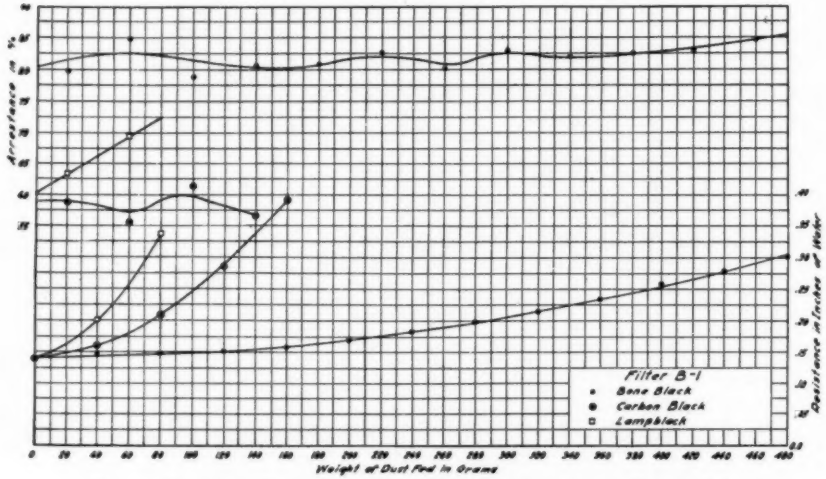

FIG. 7. EFFECT OF DIFFERENT CARBONS ON PERFORMANCE OF FILTER *B*-1. DUST FEED, 40 G PER HOUR

lampblacks place on many filters lead to the selection of a dust mixture for rating air filters for railroad applications [2] which contained only 20 per cent of lampblack instead of the 50 per cent required in the code. This change in the dust specification minimized the difficulty but did not eliminate it.

In order to find out more about the different grades of carbons and their effect on filter arrestance, four materials were selected and a study was made of the effect of these materials on the performance characteristics of the four types of filters. The materials were as follows: Lampblack, Germantown Eagle Brand, manufactured by L. Martin and Co.; Carbon Black, No. 5 grade, manufactured by Binney and Smith Co., New York City; Bone Black; No. 7 Cosmic Black, manufactured by American Agricultural Chemical Co., Detroit, Mich.; Soot, removed from the flues of an oil burning installation in which there was incomplete combustion. These materials were all screened

[2] Loc. Cit. See Note 1.

through a 100-mesh screen. The analysis is shown in Table 2. The points to note are that all dusts are made up of very fine particles, the largest per cent in each case being less than 5 microns in diameter. The densities, however, as shown in Table 4, are very different, the bone black being over three times as heavy as any of the other three. The bone black was more granular and did not agglomerate as did the other carbon dusts.

The results for the performance tests of the four filters when using each of the carbon dusts are shown in Table 4 and in Figs. 6, 7, 8 and 9 for Filters *A*, *B*-1, *C* and *D*, respectively. The results of these tests show that the arrestances for all four filters are in the order of carbon black, lowest; lampblack, second; and bone black, highest. As to the life and dust holding capacity, lampblack resulted in the poorest performance values, carbon black next, and bone black best, with the one exception of Filter *A*, in which case the dust holding capacities were about the same for carbon black and lampblack. For each case the life and dust holding capacity when using bone black were decidedly longer than for the other types of carbons. In the case of soot the tests were made for short periods on Filters *B*-1, *C*, and *D*. The arrestance of the soot was closer to that of carbon black than to that of lampblack. The density of the soot used was less than that of either the lampblack or the carbon black; therefore the life and dust holding capacity probably would be less for this material if complete tests were run. In making this investigation of different types of carbon dust the results obtained from the lampblack were very different from those obtained from the same brand of lampblack in previous tests on identical filters. The arrestances were very much higher and the life of the filter was somewhat shorter. For

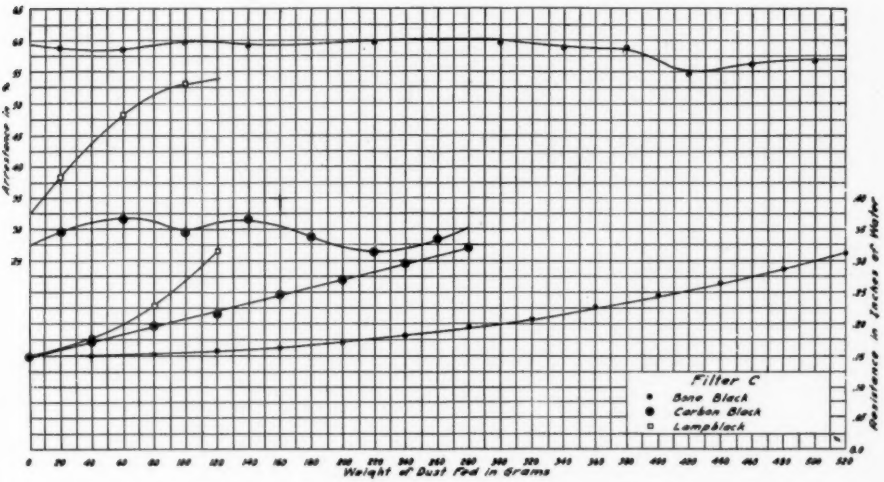

FIG. 8. EFFECT OF DIFFERENT CARBONS ON PERFORMANCE OF FILTER *C*. DUST FEED, 40 G PER HOUR

this reason an investigation was made to determine the variations in lampblack samples of the same brand.

Table 5 shows the results of tests on Filters *A*, *B*-1, *C* and *D* when using the lampblack of the previous series and that of the present series of tests.

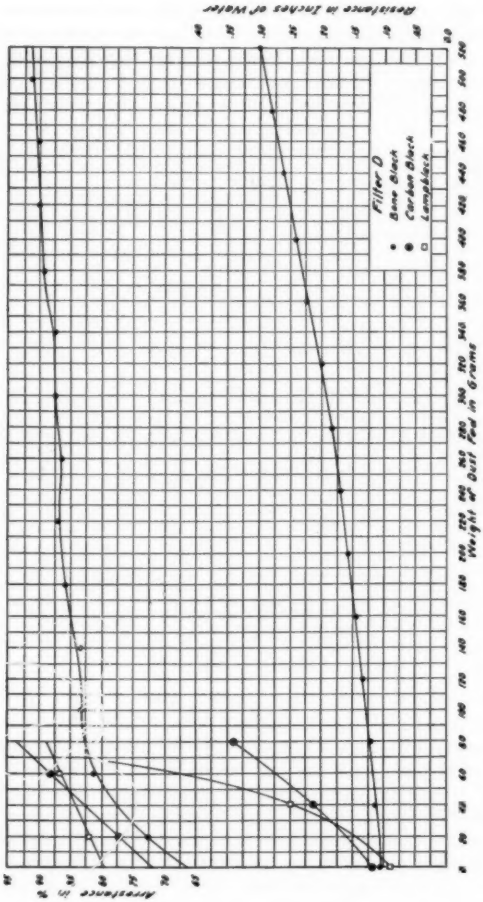

FIG. 9. EFFECT OF DIFFERENT CARBONS ON PERFORMANCE OF FILTER *D*. DUST FEED, 40 G PER HOUR

In all cases the dust arrestance and the rise in air resistance across the filter per hour when using the dust of the second series were much higher than when using the dust of the first series. It was thought at first that the relative humidity of the air at the time of test might have had some effect on the arrestance values. However, a comparison of these humidities, as taken at the time of test and shown in Table 5, showed no relationship between

humidity and arrestance. It should be noted that when tests were repeated using the same filter and lampblack from the same lot the values were substantially uniform. In each series of tests enough lampblack was purchased at the beginning of the series for the entire investigation so that the test values of a series are comparable within themselves, but the results for the different series may not be strictly comparable even though the same brand of lampblack was used.

After it was discovered that a wide range in test values might result from the same lampblack purchased at different periods, samples of the same brand were selected from as many sources as possible and a series of tests was

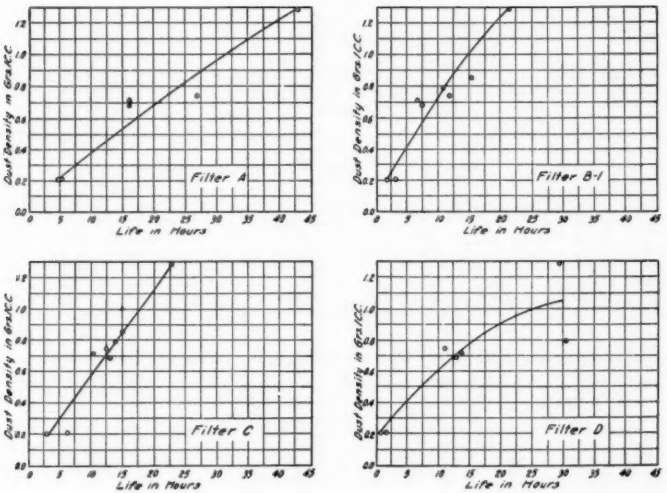

FIG. 10. EFFECT OF DUST DENSITY ON FILTER LIFE. DUST FEED, 40 G PER HOUR

run on Filter *B*-1, the results of which are shown in Table 7. Test No. 114, the last one shown in this table, was taken from the first series. This dust had the lowest density, gave the lowest arrestance value, had the lowest resistance rise per hour, and the longest filter life. The arrestance values for the lampblack of later selection ranged from 46.7 to 82.4 per cent. There does not seem to be a strict correlation between density of lampblack and arrestance values.

From the investigation thus far made on carbon it appears that lampblack and carbon black are most nearly related to soot. Soot probably corresponds reasonably well to the carbon found in the air. Bone black does not have the required characteristics. The variations in test results for lampblack of the same brand but selected from different lots is so great that further investigation is necessary in order to select a standard carbon for test purposes. This part of the investigation is now being continued.

TABLE 5—ARRESTANCE[a], RESISTANCE RISE[a], AND RELATIVE HUMIDITY DURING TEST, FOR FILTERS, *A*, *B*-1, *C*, AND *D* ON TWO LOTS OF LAMPBLACK[b]

DUST	FILTER *A*			FILTER *B*-1			FILTER *C*			FILTER *D*		
	Arr.	Resistance Rise In. Water/Hour	Per Cent R.H.	Arr.	Resistance Rise In. Water/Hour	Per Cent R.H.	Arr.	Resistance Rise In. Water/Hour	Per Cent R.H.	Arr.	Resistance Rise In. Water/Hour	Per Cent R.H.
Performance with first lot Lampblack	21.7	0.001	46	27.8	0.008	46	23.3 17.1	0.007 0.007	46 42	74.1	0.096	..
Performance with second lot Lampblack	74.3	0.013	45	65.2 56.8 63.2	0.098 .... 0.084	40 22 24	45.4 37.7 36.8	0.055 0.074 0.079	22 26 40	82.5	0.175	29

[a] Based on 3-hour average.
[b] Both lots same brand from same manufacturer.

TABLE 6—RELATION BETWEEN DUST DENSITY AND FILTER PERFORMANCE[a]
(*Dust Feed* 40 *g per hour. Face air velocity* 300 *fpm*)

TYPE OF DUST	DUST DENSITY		DIAMETER PARTICLE, MICRONS, WITH AVG. VOL.	LIFE IN HOURS				DUST HOLDING CAPACITY IN GRAMS				ARRESTANCE IN PER CENT			
	Grams/ cc	Pounds/ cu ft		*A*	*B*-1	*C*	*D*	*A*	*B*-1	*C*	*D*	*A*	*B*-1	*C*	*D*
Soot	0.135	8.5	..	..	..	..	..	..	..	..	..	..	58.9	29.7	66.9
Lampblack	0.203	12.7	7.2	4.8	1.8	2.8	0.9	146.6	45.2	50.8	29.5	75.9	66.0	45.4	82.0
Carbon black	0.210	13.2	7.0	5.1	3.1	6.2	1.5	145.8	71.0	72.4	48.5	71.4	58.2	29.4	80.9
Pocahontas 200–325	0.686	42.8	22.3	16.0	7.4	12.9	12.7	604.1	277.4	441.8	464.0	94.4	93.5	84.9	91.8
Pocahontas 325+	0.718	44.8	10.3	16.0	6.7	10.3	13.5	585.6	249.8	325.5	511.0	91.4	93.2	77.7	94.5
Boneblack	0.741	46.3	5.9	26.9	11.8	12.5	10.8	838.4	386.0	292.4	365.4	78.7	81.8	58.2	84.7
Illinois Fly-Ash 325+	0.795	49.7	9.7	..	10.8	13.9	30.4	..	415.9	457.7	1152.8	..	96.3	81.6	94.6
Illinois Fly-Ash 200–325	0.857	53.5	20.1	..	15.3	15.0	..	..	588.9	497.0	..	..	96.2	83.3	90.6
Cottrell Ash	1.285	80.4	6.0	43.0[b]	21.4	22.6	29.2	1590.0[b]	815.1	709.4	1021.6	92.3	94.7	78.4	87.7

[a] All values based on a resistance rise of 0.15 in. of water across filter.
[b] Estimated value.

TABLE 7—COMPARISON OF LAMPBLACKS[a] ON FILTER *B*-1
(*Dust feed* 40 *g per hour. Face air velocity* 300 *fpm*)

TEST NO.	AVG. ARRES. FOR 2 HOURS	LIFE[b] IN HOURS	AVG. RESIS.[c] RISE PER HOUR	DUST DENSITY	DUST SAMPLE	SOURCE
256	78.8	1.8	0.088	0.183	1	A
257	74.0	1.8	0.089	0.213	2	
258	78.4	1.9	0.084	0.203	3	
259	75.6	2.3	0.063	0.206	4	
260	71.1	2.1	0.074	0.209	6	B
252	82.4	1.9	0.087	Approx. same as 1, 2, 3, 4	..	A
211	64.4	1.9	0.084		..	
212	56.8	..	..		..	
217	66.2	1.8	0.098		..	
255	58.4	1.5	0.049	0.265	5	C
261	46.7	2.3	0.066	0.238	7	D
114	28.1	Approx. 4.5	0.024	0.156	..	

[a] All lampblacks same brand and same manufacturer but from different supply houses.
[b] Based on final resistance of 0.30 in. of water across filter.
[c] Based on 2-hour average.

DUST DENSITY AND FILTER PERFORMANCE

The relation between filter performance and density of dust is shown by the tabulated results in Table 6 and by the curves of Fig. 10. From the table it will be noted that the density of soot, the lightest dust, is about one-tenth that of Cottrell ash, the heaviest dust; the lampblack and carbon black used were of substantially the same density, about one-sixth that of Cottrell ash. The finely screened Pocahontas ash was slightly heavier than the coarsely screened ash, but the reverse was true of the Illinois fly-ash. In all cases

TABLE 8—DUST HOLDING CAPACITIES[a] IN CUBIC CENTIMETERS

TYPE DUST	DENSITY (JOLTED) GRAMS/CC	RATE OF FEED[b] IN CC/HOUR	DUST HOLDING CAP. IN CC			
			Filter *A*	Filter *B*-1	Filter *C*	Filter *D*
Lampblack	0.203	199	722	222	251	146
Carbon Black	0.210	190	693	338	345	231
Pocahontas Ash 200–325 mesh	0.686	58	881	404	644	677
Pocahontas Ash 325+mesh	0.718	56	816	348	453	711
Boneblack	0.741	54	1130	522	395	492
Illinois Fly-Ash 325+mesh	0.795	50	..	522	575	1450
Illinois Fly-Ash 200–325 mesh	0.857	47	..	688	580	..
Cottrell Ash	1.285	31	..	635	552	795

[a] Based on a resistance rise of 0.15 in. water across filter.
[b] Based on a weight of 40 g. per hour.

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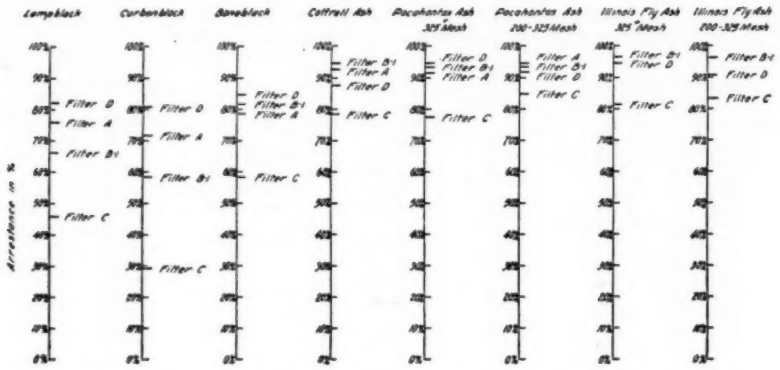

FIG. 11. COMPARATIVE FILTER ARRESTANCES FOR DIFFERENT DUSTS. DUST FEED, 40 G PER HOUR

Illinois fly-ash was slightly heavier than the Pocahontas ash, but the Cottrell ash was 50 per cent heavier than either of the other two for the same volume.

The life in hours and dust holding capacity in grams were both determined on a basis of 0.15 in. of water increase in resistance across the filter. The graphical relation between dust holding capacity and dust density is similar to that for life and dust density, but lack of space does not permit its inclusion. In all cases the life and dust holding capacities increase with increased density. This is to be expected since each filter has a definite volumetric dust holding capacity, and for the heavier dust the volumetric feed is in inverse relation to the density. The arrestance values, as shown in Table 6, do not bear the same direct relation to the density of the dust, as there is only a slight positive correlation between the two. For Filters *A*, *B*-1, and *C* the arrestance values are considerably lower for soot, lampblack and carbon black, and for Filter *D* they are slightly lower for these dusts than for the others. There

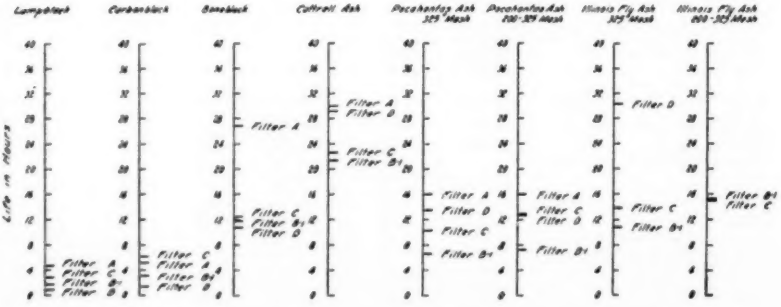

FIG. 12. COMPARATIVE FILTER LIFE FOR DIFFERENT DUSTS. DUST FEED, 40 G PER HOUR

are apparently other factors than dust density which affect the arrestance values. This is evident by the variations in results for the different lampblacks, even though the densities were substantially the same (See Table 7).

Both the life and dust holding capacity of all filters show a decided increase as the density of the dust is increased, with a constant weight of dust feed per hour. If, however, the dust holding capacities are based on volume of dust feed, there is a much closer relation between dust holding capacity and different types of dust for the same filter. This relation is shown in Table 8. In general, lampblack and carbon black show the lowest volumetric capacities for each filter. For Filters *A*, *B*-1, and *C* the coarsely screened Pocahontas

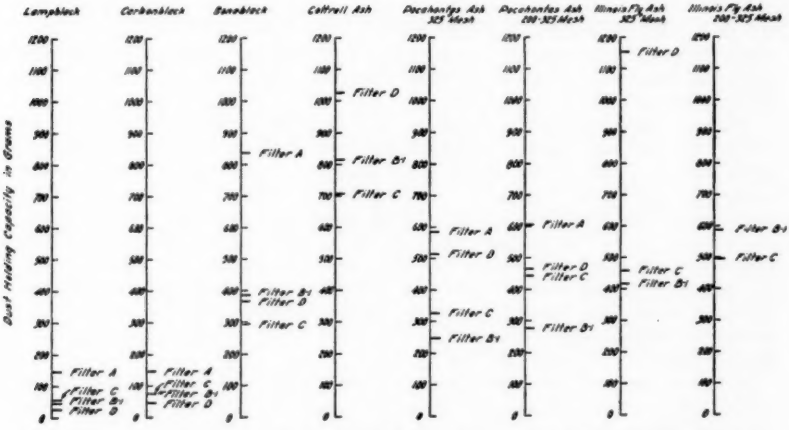

FIG. 13. COMPARATIVE FILTER DUST HOLDING CAPACITIES FOR DIFFERENT DUSTS. DUST FEED, 40 G PER HOUR

and Illinois fly-ash show slightly greater capacities than for finer screened ashes, but for Filter *D* the results are in the inverse order.

In rating a filter for life and dust holding capacity it seems evident that a dust should be selected which not only has the proper grain size, but also one which has a density which corresponds reasonably well with the density of dust to be found in the air. Of the coal ashes used, it appears that Pocahontas should have the highest rating and Cottrell ash the lowest rating as to this quality.

GENERAL COMPARISONS OF FILTER PERFORMANCE

The relative values for arrestance, life in hours, and dust holding capacity in grams, for the four types of filters tested with different types of dust are shown graphically in the diagrams of Figs. 11, 12 and 13. The life in hours and dust holding capacity in grams were based on an air resistance rise

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across the filter of 0.15 in. of water. The average initial test resistances of the filters at 300 fpm face velocity air were as follows:

A—0.233 in. water
B-1—0.145 in. water
C—0.149 in. water
D—0.101 in. water.

All filters were 2 in. in thickness excepting *A* which was 4 in. in thickness. It is recognized that a resistance rise of 0.15 in. of water is less than that specified in the code or than would be expected in practical operation of the filters. This limit was used, however, because of the excessive time that would have been required to run all tests up to a final filter resistance of 0.4 or 0.5 in. of water. The low limits of resistance used should not have any effect on the comparative results for a given filter when using different types of dust, but might have some effect on the comparative values for different filters on the same dusts.

The arrestance values from Fig. 11 show a wider spread for the values of different filters when using lampblack and carbon black than when using the other types of dust. All filters show satisfactory arrestance values when using any of the ash materials. Filter life, as shown in the diagram of Fig. 12, is very short in all cases when using lampblack and carbon black. In general, it is the longest when using Cottrell ash. The dust holding capacities in grams as shown in Fig. 13 follow substantially the same relation as the values giving life in hours. In general, as previously pointed out, the lighter weight dusts show low dust holding capacity and life due to the volumetric limitations of the filters. It should be noted that no one type of filter is best for all types of dust.

Table 9 shows the life comparison of three filters, *B*-1, *C*, and *D*, when tested with a given mixture of dust and compared with the weighted life for the same filters tested with the individual dusts of the mixture. The figures in the last column show that there is a low correlation between the weighted and actual test values, the weighted life being the greatest in all cases. In other words, when two dusts are used in a mixture, one of which gives a short life to the filter and the other a long life, the short life dust has a greater effect on the life of the filter than would be expected from its actual percentage in the mixture. This is in contrast to the results obtained by weighting the values for filter arrestances of individual dusts and comparing them to the arrestance values for the combined dust. In this case there was a very high correlation of results, as reported in the previous paper.

Figs, 14, 15, 16, and 18 show the performance curves of Filters *A*, *B*-1, *C* and *D* respectively when using two different dust mixtures, the first consisting of 50 per cent Pocahontas ash and 50 per cent lampblack as described by the Code, and the second containing 50 per cent Pocahontas ash, 20 per cent lampblack, 20 per cent Illinois fly-ash, and 10 per cent Fuller's Earth. All filters show a marked reduction in arrestance and life when using the Standard Code dust as compared with the 50-20-20-10 mixture. Fig. 17 shows the performance curves for the same filters when using Cottrell ash. This ash has been used unmixed, in some cases, as a test dust. From a comparison of the curves for the different dust mixtures and Cottrell ash it will be noted that in all cases the filter life was longest when using the Cottrell ash. For

Filters *A*, *B*-1, and *C* the arrestances were higher when using the Cottrell ash than for either of the two dust mixtures. For Filter *D* the arrestance was lower when using the Cottrell ash.

While it is apparent that a dust mixture containing 50 per cent of lampblack as called for in the Standard Code probably penalizes the performance of most filters, it is also evident that Cottrell ash is not an ideal dust to use for rating air filters because its high density results in performance characteristics better than those to be expected in practice. The 50-20-20-10 mixture seems to be preferable to the other two test dusts used, but there still is a question

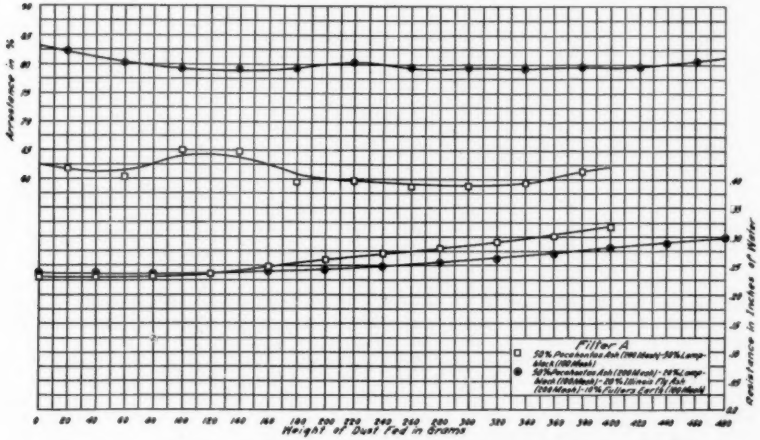

FIG. 14. PERFORMANCE CURVES FOR FILTER *A* ON TWO DUST MIXTURES. DUST FEED, 40 G PER HOUR

as to whether this mixture has all of the qualities for a practical dust mixture to be accepted as a standard for test purposes. It seems probable that a satisfactory standard dust might be made up of not more than two or three components. These components should be of low density. Further investigation should be made in order to select a carbon dust, as well as ash dust, which can be duplicated with reasonable accuracy.

CONCLUSIONS AND RECOMMENDATIONS

A summary of the conclusions drawn from various sections of this discussion are as follows:

(1) In rating a filter it is not practical to duplicate the extremely low dust concentrations met in practice. Tests have shown that rates of dust feed in excess of that specified by the Standard Test Code give filter arrestances which are comparable to those obtained at the low rates found in practice. It seems desirable in revising the Air Filter Test Code to increase the rate of dust feed.

(2) In general, within the range obtainable by screening, dusts of large particle size are easier to filter from the air than similar dusts of small particle size. From

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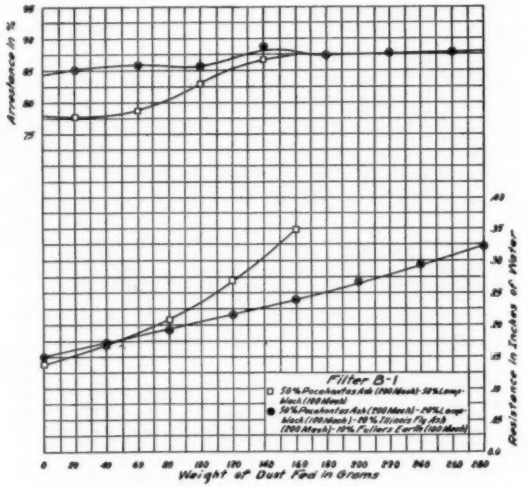

FIG. 15. PERFORMANCE CURVES FOR FILTER *B*-1 ON TWO DUST MIXTURES. DUST FEED, 40 G PER HOUR

the results thus far obtained it would seem preferable to screen all ash dusts through a 325-mesh screen. Further research should be made in an attempt to determine the effect on filter performance of dusts finer than those obtainable by screening.

(3) Lampblack and carbon black both have the effect of reducing filter life and dust holding capacity when used as a test dust. Both of these carbons approximate soot fairly closely, as far as their effect on filter arrestance is concerned. It is evident that there is a wide variation in the properties of these materials, even though the dust appears to be of uniform specifications. The carbon used in a standard test dust should be of low density and should be capable of duplication as to its properties

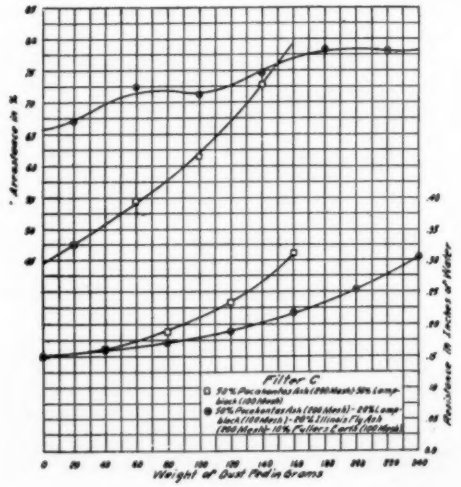

FIG. 16. PERFORMANCE CURVES FOR FILTER *C* ON TWO DUST MIXTURES. DUST FEED, 40 G PER HOUR

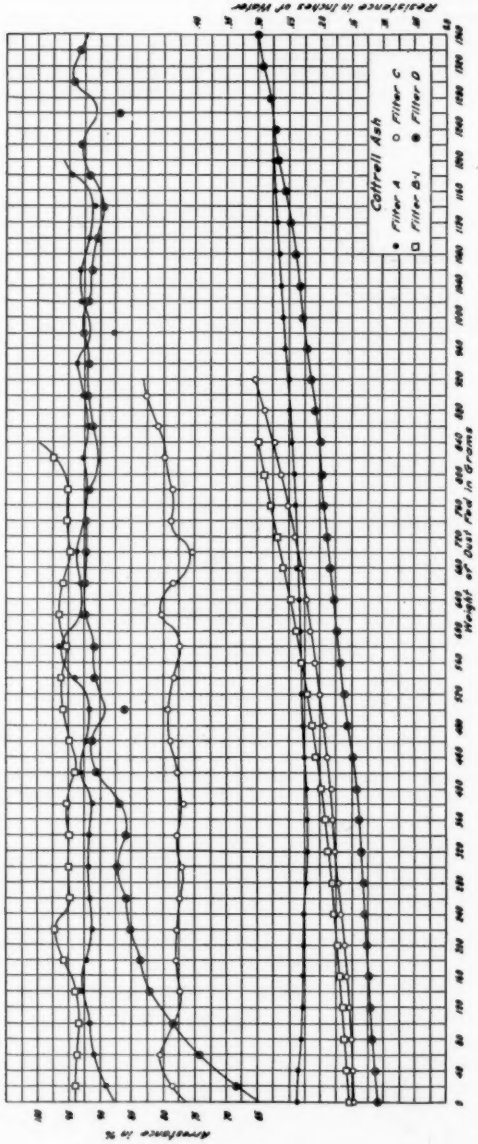

FIG. 17. PERFORMANCE CURVES FOR FILTERS *A*, *B*-1 *C* AND *D* ON COTTRELL ASH.
DUST FEED, 40 G PER HOUR

affecting filter performance. Further investigation should be made in an effort to standardize on a carbon for the standard dust.

(4) In general, filter life and dust holding capacity increase with increased dust density because of the constant volumetric capacity of a filter. There is a slight positive correlation between filter arrestance and increasing dust density. For these reasons dust density is an important factor in rating the performance of a filter on the weight basis.

(5) The present Standard Code dust of 50 per cent lampblack and 50 per cent Pocahontas ash probably penalizes the performance of most filters. On the other

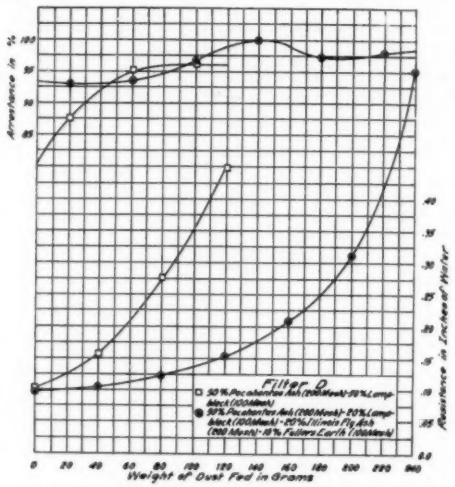
Arrestance in %
Resistance in Inches of Water
Weight of Dust Fed in Grams
Filter D
50% Pocahontas Ash (100 Mesh)-50% Lampblack (100 Mesh)
50% Pocahontas Ash (200 Mesh)-50% Lampblack (100 Mesh)
50% Pocahontas Ash (200 Mesh)-20% Illinois Fly Ash (200 Mesh)-30% Fullers Earth (100 Mesh)

FIG. 18. PERFORMANCE CURVES FOR FILTER *D* ON TWO DUST MIXTURES. DUST FEED, 40 G PER HOUR

hand, the test dust consisting of straight Cottrell ash results in performance characteristics better than those obtainable in practice. It seems probable that a satisfactory standard dust may be made up of not more than two or three low density components.

These tests were made for the purpose of determining possible improvements in the Standard ASHVE Test Code, and not for the purpose of comparing the different filters used. To compare the merits of the different types

TABLE 9—COMPARISON OF ACTUAL LIFE OF FILTER WITH WEIGHTED LIFE FOR 50–50 MIXTURE

FILTER	ACTUAL LIFE[a] IN HOURS	WEIGHTED LIFE IN HOURS	PER CENT ACTUAL LIFE OF WEIGHTED LIFE
B-1	3.4	4.6	73.9
C	3.9	6.8	57.3
D	2.2	8.1	27.2

[a] Based on final resistance of 0.30 in. of water across the filter.

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of filters would require a much more extended series of tests, and any attempt to make comparisons from the data presented here would be misleading. The filters used were selected because they were typical of their classifications, and it should not be construed that they are superior to other filters within their classifications.

ACKNOWLEDGMENT

The authors wish to acknowledge the cooperation of F. C. Houghten, Director, ASHVE Research Laboratory, and the Technical Advisory Committee on Air Cleaning and Atmospheric Impurities, of which H. C. Murphy is Chairman. It is also gratifying to note that the manufacturers of the various types of filters used were generous in the supply of any materials requested, and gave their full cooperation to make the program possible.

DISCUSSION

ARTHUR NUTTING[3] (WRITTEN): This series of tests is especially valuable because it emphasizes the fact that the results of filter tests are probably affected more by the size, character, and weight of the dust used than by any other factor. Careful consideration should, therefore, be given any proposed change in the ASHVE filter test Code which would contemplate a change in the kind of dust used.

The purpose of a code is to permit the rating of an air filter in such a manner that its field performance can be predicted. Ratings established under a fairly devised code will reflect the comparative worth and value of various types of filters as air cleaning devices. These ratings should be a function of the filter efficiency and dust-holding capacity, and in the last analysis will be proportionate to the degree of air cleaning obtained and the cost of accomplishing that cleaning.

As mentioned above, the dust used in testing affects the rating more than any other factor and Fig. 11, I believe, indicates this point. The tests made with 200-325-mesh Pocahontas ash show an arrestance of from 85 per cent to 95 per cent between the highest and lowest filter. The test made with carbon black shows a spread of from 29 per cent to 81 per cent, a total of 52 percentage points, or more than 5 times as much as in the previous example. In these tests the filters selected cover a broad range of types, and those of lower arrestance and life are not sold as the same class of equipment as those of higher rating. One would, therefore, expect the rating between the highest and lowest to be quite different and to correspond with the service which we know is obtained with these two types in practical installations.

Cottrell ash and like dusts are, it is believed, too different from most atmospheric impurities to justify their use. Filters which are 20 per cent apart in arrestance with ASHVE dust would probably be only 2 per cent or 3 per cent apart with Cottrell dust, and this 2 per cent or 3 per cent difference is very little more than the usual experimental error in testing filters.

If the problem in the air filtration art had been based on the removal of dusts of a nature similar to Cottrell or Pocahontas ash then the design and practice of air filtration would have been greatly simplified throughout the entire existence of the air filter industry. It is a fact, however, that the most troublesome atmospheric impurities and those most prevalent are carbons, which are often quite similar to lampblack or carbon black, and the elimination of this type of solids is an entirely different

[3] Chief Engineer, American Air Filter Co., Inc., Louisville, Ky.

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problem and one more difficult than the elimination of ashes. The fineness, greasiness, bulkiness and general character of the carbonaceous impurities not only complicate the air filtration problem, but these qualities, in addition to the carbon's high discoloration factor, impose a need for a higher degree of air cleaning than if we worked with dusts of a far different character.

In conclusion, I believe it necessary that any test dust should be composed of a high percentage of carbons and that the dust as a whole be made of as few components as possible, and that these components be dusts that can be procured by any one at any time.

W. J. OTTINGER:[4] I should like to ask whether the machine has a tendency to segregate dusts of various kinds?

PROFESSOR ROWLEY: All of the dust in the mixture is picked up by the air stream and carried into the filter without separation. There may be some separation of the dust by gravity as it passes through the horizontal duct to the filter, but this is very slight and all of the dust which settles in the duct is cleaned out by an air stream and passed into the filter before final measurements are taken. Perhaps one improvement of the present air filter test code would be to use a vertical instead of a horizontal duct and thus prevent any settling of the dust from the air stream passing into the filter.

J. W. MAY: If, in the future, the code should recommend higher concentrations, up to 40 g per hour; what is his opinion regarding the advantages to be derived from having the code specify a permissible spread, because some types of filters have such low dust-holding capacities that 40 g per hour would not give enough points to plot an arrestance curve.

PROFESSOR ROWLEY: In case a filter had a very low dust-holding capacity it might be desirable to reduce the rate of dust feed rather than to increase it above the present code requirements. Most filters, however, have sufficient dust holding capacity to make the higher rates of feed practical for test purposes.

G. W. PENNEY: I think this paper gives a valuable illustration of the range in efficiencies obtained using different test dusts, even though tests were not made using any dusts as fine as would be representative of the particles in most city air. However, the range of efficiencies obtained for Filter *C* of 84.9 per cent for Pocahontas ash down to 29.7 per cent for soot at least gives an illustration of the importance of the type of dust used for testing.

If, in addition to this range in particle size of test dust, a dust could have been included which would be fine enough to represent the smoke particles which represent the most of the contamination in the air in residential sections of many cities, and if the efficiency could have been tested both by weight and by the blackness method described by Mr. Dill of the Bureau of Standards at the summer convention, I believe we would have a really startling range in efficiencies on a given filter.

I think that tests such as these should convince anyone of the limitation of laboratory tests unless the dust used in the test accurately represents the material to be removed from the air in service and unless the method of measuring the efficiency represents the basis on which the performance of the filter will be judged in operation. I believe the question of whether the weight basis of measuring efficiency or the blackness method described by Mr. Dill most accurately represents requirements in service deserves serious consideration, for the two methods will, in general, give quite different efficiencies, and in many conditions of service we are primarily interested in discoloration of walls, draperies, etc., rather than weight of dust removed.

[4] The Air Conditioning Supply Co., Cleveland, Ohio.

PROFESSOR ROWLEY: The question of the comparative results of filter tests by different test methods is important as several laboratories are rating filters by different methods. The relative efficiencies and other qualities of filters will apparently be different when rated by different methods.

E. C. EVANS: I would like to ask one question. In your conclusion 4, I presume it is your idea that filter life and dust holding are identical?

PROFESSOR ROWLEY: Filter life is based on the total amount of dust fed to the filter and dust-holding capacity is based on the amount of dust retained by the filter.

H. C. MURPHY: The authors have developed additional information which is required in the development of the alternate testing method of testing air filters which our committee has in hand. There are in addition much data which will be of value in the measurement of atmospheric impurities. The information regarding rates of dust feed, for instance, is particularly informative, as up to the present time it has been thought that the rate of feed specified in the present air filter test code was excessive. In European tests, the dust feeding continued over several days with concentrations approximating that of the normal air.

In view of the data developed in this investigation, it seems desirable that the committee recommend an increase in the rate of dust feed in the forthcoming revision of the Air Filter Test Code. Many of the conclusions confirm the premises on which the present Society's code is based.

It was realized, for instance, when the present code was developed, that soot, lampblack and the like, were among the most difficult of all materials to remove from an air stream. It was for this very reason, however, that these materials were selected as major constituents of the standard test dust.

Now, it would be obvious that if we used *BB* shot or marbles against such materials, all filters would be 100 per cent efficient, and the thought in using a very difficult dust such as soot was to establish a difference between different filters.

It is entirely possible, as Professor Rowley points out, that lampblack as used in the present standard test dust, penalizes the arrestance values of most filters; but in my own opinion it would be a mistake to substitute a material which is too readily removed or which does not conform fairly closely to the material found in ordinary city air.

The present code has in its preamble this note: "This Code is for the Laboratory investigation and rating of devices used in general ventilation work for the sole purpose of removing solid impurities from the air. It is not intended for use in the rating of cyclones, dust separators or for devices used in the field of industrial hygiene. Ratings established under this code are not to be confused with operating efficiencies in actual service. The dust concentration, the nature of the dust, the relative humidity, and many other factors have a definite bearing upon results obtained in actual service."

In other words, this code is intended simply as a yardstick, and any attempt to use it for measuring the performance of filters under operating conditions will result in confusion.

PROFESSOR ROWLEY: Both Mr. Murphy and Mr. Nutting refer to the desirability of retaining a high percentage of carbon dust in the synthetic dust used for testing air filters. There are many unknown factors concerning the carbon black specified, and there are many different qualities which will answer the requirements of the Standard Code and give different test results. It is also questionable as to how much carbon black the average air filter will be subjected to. This is a problem which needs much more study and it may be advisable to test filters with different dust mixtures to get comparative ratings which will meet field requirements.

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R. S. DILL:[5] I found what Professor Rowley had to say about dust-holding capacities very interesting, and it should be of interest to anybody considering air filter tests. Professor Rowley and ourselves seem to have approached this subject with somewhat different concepts. For instance, we considered that any dust so large that you could separate it by screening, was too large for consideration in air filter tests. For this reason, perhaps, I do not understand all there is in the paper; but Professor Rowley has promised to come to Washington and further explain it.

His conclusion that the rate of dust feed during air filter tests, at least between rather wide limits, does not affect the efficiency found, corresponds with our experience. We have never tried to test with lampblack because our microscopist refused to try to give a figure on its size. He said we could not tell what its condition was when it was suspended in the air.

JOHN JAMES: Would it be possible to develop some form of alternative test code to ascertain the efficiency of filters in the field for acceptance tests? In the Code of Minimum Requirements of Comfort Air Conditioning, which the Society adopted last January, there is a statement on air quality and air impurity, which indicates that air filtering devices shall be capable of removing 95 per cent by count of all dust particles over 10 μ in size from all air delivered to the enclosure. In the field it is a question as to how one can actually meet this specification in the code, and there has been suggested, in addition to laboratory methods of testing air filters, a simple testing technique consisting of sampling devices placed on the entering and leaving side of the filter, which could be used after the equipment has been installed. I wonder whether Professor Rowley, with his long experience in air filtering work, would care to comment on the practicability of using some form of field test for measuring the efficiency of a filter.

PROFESSOR ROWLEY: Referring to the last question brought up by Mr. James, I believe that there is no standard method acceptable to all concerned for measuring the amount of dust in the air. A practical method of measuring the amount of dust in the air seems to be one of the big problems in studying air purity.

I agree with Mr. James that a simplified test method should be developed which could be correlated with the standard code and which would be practical to use in the field. A reliable dust count method might be acceptable for determining the efficiencies of filters in the field. There are several dust count methods now in use but the results from the different methods do not check and in many cases there is no consistent correlation factor. We have been working on an impinger type of dust counter which gives very high dust counts and which appears to have practical application for field work. The discoloration method which has been used for rating filters is rather simple and easy to apply, but it seems doubtful that consistent results could be obtained for different types of dust. For instance a small amount of carbon dust would give a very high discoloration factor or obstruction to the passage of light through a filter paper as compared with the same amount of some light colored dust. If this method was to be used for rating filters some consideration would have to be given to the kind of dust encountered by the fitlers.

I would like to add a word in regard to reducing the carbon content of standard dust from 50 per cent to some lower value such as 20 per cent. Since the density of lampblack used is very much less than the coal ash dust specified, a mixture of 50 per cent of lampblack to 50 per cent of coal ash by weight gives a ratio of 4 or 5 to 1 by volume. This seems to be entirely too much carbon for any practical conditions and filters tested by this kind of dust might give entirely different results than when used for dust as found in practice.

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[5] National Bureau of Standards, Washington, D. C.

J. H. FERGUSON: The Bureau of Mines was doing considerable work in dust-laden atmospheres, and they do use electric precipitators, referred to by Professor Rowley. Also, I think their method of manufacturing dust-laden atmospheres is much simpler than the method used here. They use a calibrated tube with a nozzle remaining stationary, and the contents of the tube are sent toward the nozzle with a telechron motor at a non-variable rate. It seems to be a very accurate method.

PROFESSOR ROWLEY: I believe that any method by which the dust could be fed continuously at a uniform rate into the air without separation would be satisfactory. We have always found it difficult to feed the dust through a small tube at a uniform pre-determined rate.

PROF. AXEL MARIN: There has been one question raised, and I would like to stress it just a bit more, and that is this question of getting the same sample of dust; even though you get it from the same source, it differs rather widely. I have been a bit concerned about it in this respect; say Professor Rowley should run a series of filter tests on one filter with a different dust, and some other individual gets what he believes to be the same dust from this particular concern, and his filtering efficiency might be 29 per cent and Professor Rowley's might be 85 per cent. There would be considerable argument. The question then is what you would consider the minimum number of tests that should be run in order that you might have what might be termed a fair or representative comparison between, say, one set of filters, with some other set, and perhaps that leads us around to the point that the Society should, inasmuch as they have specified a minimum code requriement, naturally now do something about broadening these samples to be used in the rating of these filters.

PROFESSOR ROWLEY: It seems to me that the minimum number of tests required will depend largely upon normal variations. If we consider variations from 29 to 85 per cent, obviously it would be difficult to get any satisfactory rating value which could be used with any degree of certainty. I believe that it will be possible to ultimately find a standard dust mixture which will give much more consistent results and which will be satisfactory.

J. W. KREUTTNER: I wanted to ask whether consideration of the humidity and temperature of the air, and its effect on the filtering medium, the microscopic medium, was given. What about relative humidity?

PROFESSOR ROWLEY: The tests reported in this paper were run at laboratory temperatures which varied from 70 to 85 deg. A series of tests are now being made in which cold outside air is drawn through the filter. The temperature will apparently have some bearing on the efficiency. Relative humidities have been taken but so far as we have been able to determine the normal range found in laboratory air has no effect on efficiency. This was one of the first points checked in trying to find the cause of the wide variation in results with different samples of lampblack, but the analysis showed that relative humidity made practically no difference.

MR. KREUTTNER: The high humidities do not affect the filtering media of a hygroscopic nature?

PROFESSOR ROWLEY: It may be that very relative humidities will affect filters made of hygroscopic material. This point requires more test data.

MR. KREUTTNER: Do you not have as much of the high humidity in the normal atmospheric air as you do at the low? Your night temperatures are lower; your night humidities are higher.

PROFESSOR ROWLEY: We have run tests both day and night through various types of weather, and therefore have had quite a range of relative humidities. The

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variation, however, in tests due to type of dusts and different samples of same kind of dust have been so great that any effect of humidity was not apparent.

MR. DILL: During our tests at the Bureau of Mines, which were made with air from the outside of the building, on dry days and on rainy days, we did not find any difference made by the humidity of the air on the efficiency of the filter. I do not know whether anybody found different results or not; but that goes for sheeps wool, cotton, steel wool, and other things you might expect an effect on.

PROFESSOR ROWLEY: Confirming Mr. Dill's experience as to humidity, I might say that we have kept the relative humidity record for practically all of our tests and have not been able to detect that relative humidity makes any difference in the efficiency.

H. J. ROSE: I understand that the code specifies a dust mixture containing 50 per cent Pocahontas ash. Is this ash specifically described? That is as to seam, mine, and type of plant in which the Pocahontas coal was burned?

PROFESSOR ROWLEY: No it is not. There is no specific description excepting it is screened through a 200-mesh screen.

MR. ROSE: Pocahontas coal is produced not merely from one seam but from four different and distinct numbered seams, known as Pocahontas No. 3, 4, 5 and 6 seams, and the fusion point of ash of Pocahontas coal from different sources has a surprising variation, ranging from high to very low. Also, it is quite possible for lump Pocahontas to have a different ash composition and characteristics than the ash from slack coal from the same mine. Therefore, a very definite source of Pocahontas ash or other coal ash should be specified; otherwise, there may be some rather large and annoying variations due to that source.

It certainly is difficult to find standard dusts that are really suitable for testing of this sort. With regard to the large variations found in carbon dusts, I am acquainted with a research group which has been doing a great deal of work in producing carbon black under rigidly controlled conditions.

PROFESSOR ROWLEY: We are at present investigating ash from Pocahontas coal taken from different seams and do find a variation in density. It appears that there will be a definite relation between filter performance and density of dust. It may be possible to eliminate this variation by selecting ash within a given density range.

MR. MURPHY: When the problem of developing an air filter test code was first given to the committee, it appeared to be a relatively simple matter, after hearing the paper this afternoon you can see it involves many unexpected complications. It is indeed a difficult problem. The committee is developing a code for field testing, and that should be available within the next few months. This I believe will answer Mr. James' question.

J. E. RYAN:[6] Would it be of any advantage to have two separate tests with rather pure dust, one using ash dust and a second using some light fluffy dust, such as lampblack? Would it involve too much complexity to state an arrestance for example, as 85 to 50, showing how the filter behaved with each of two rather standard and pure types of dust? What do you think of that, Professor Rowley?

PROFESSOR ROWLEY: We have found that when a filter is tested with two different types of dust and then tested by using a mixture of the two dusts the efficiency when using the dust mixture is equal to the weighted efficiencies for the individual dusts. It would therefore seem reasonable to make tests with the two different types of dust as suggested by Mr. Ryan. The objection to this procedure would be the time and expense required for the tests.

[6] General Electric Co., Schenectady, N. Y.

No. 1123

EFFECT OF HEAT STORAGE AND VARIATION IN OUTDOOR TEMPERATURE AND SOLAR INTENSITY ON HEAT TRANSFER THROUGH WALLS

By J. S. ALFORD,* LYNN, MASS., J. E. RYAN,** SCHENECTADY, N. Y., AND F. O. URBAN,*** BLOOMFIELD, N. J.

INTRODUCTION

AN accurate estimate of either the heating or cooling load is essential to an economic selection of air conditioning equipment. These loads are influenced by five basic items, namely:

1. Heat transmission through walls, windows, etc., resulting from a difference between the temperature of the air on one side of the partition and the temperature of the air on the other side.
2. Radiant energy of the sun absorbed by the outer surface of an exposed wall or transmitted directly through glass partitions.
3. Sensible heat and moisture of outside air introduced to the conditioned space either by infiltration or by controlled ventilation.
4. Sensible heat and moisture of occupants.
5. Sensible heat and moisture liberated by lights, electrical equipment, gas-burning equipment and other miscellaneous sources.

The loss of heat from a structure in winter is occasioned by items 1 and 3 only. Hence these are the only ones considered in determining the maximum heating load. The other three items are heat gains, or deductions from the heat loss, and may be of importance in the design of the system and its control.

The heat gain of a structure in summer includes all five items listed. Hence, each one must be considered carefully in calculating the maximum refrigeration requirements for summer air conditioning. In certain structures heat transmission through walls and radiant solar energy transmitted directly through glass partitions account for a large share of the total cooling load. The transmission of heat through walls exposed to the sun is greatly increased

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* Turbine Engineering Dept., General Electric Co.
** Engineering General Dept., General Electric Co.
*** Air Conditioning Dept., General Electric Co.

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because of the radiant energy absorbed by the outer surface of the wall. For example, it is not uncommon for the temperature of the outer surface of a wall exposed to the sun to rise 15 to 30 F above the outdoor air temperature, and even more. Temperatures of flat roofs have been observed as high as 180 F.

A previous paper [1] summarizes the entire heat gain calculation and presents a rational method for determining the increase in heat transmission rate through a wall or roof resulting from exposure to the sun.

Another factor which has been recognized for some time as having a material bearing on the magnitude and time of occurrence of the maximum cooling load is the heat storage capacity of the wall structure. In 1930 Stevenson and Bojer[2] analyzed the transient flow of heat through refrigerator box walls, and obtained a striking agreement between calculated and test results. Two years later, Houghten, Blackshaw, Pugh and McDermott [3] investigated the flow of heat through several roof panels as affected by diurnal variations in outdoor air temperature and solar intensity, obtaining equally good agreement between analytical and experimental work. The importance of heat storage capacity upon the cooling load has been further demonstrated in reports to the Society by several authors, in which considerable discrepancies between calculated and measured cooling loads of actual air conditioned structures have been observed. In some of these cases the nature of the discrepancies gives every indication that they result from the influence of heat storage capacity of exposed walls upon the transmission of heat, and of heat storage capacity of interior partitions and furnishings upon the release of radiant energy transmitted directly through windows.

Heretofore, a consideration of the influence of heat storage capacity upon the cooling load has necessitated working from an assumed temperature of the outer surface of the wall by the method given by Houghten et al [3] or else has necessitated rough approximations.[1] The rough approximation is adequate where other elements of the heat gain, such as ventilation air, occupancy, lights, etc., account for a major portion of the total heat gain, but in other cases is undesirable. On the other hand, the outside surface temperature of a wall depends upon a large number of factors, such as:

1. The latitude.
2. The orientation of the wall.
3. The time of day and time of year.
4. The degree of cloudiness, smokiness and the humidity.
5. The nature and color of the outside surface.
6. Heat transmission and heat storage characteristics of the wall structure.
7. The temperature maintained on the inside of the wall.

Therefore, an extension of the analysis is desirable in order to base it upon the more fundamental considerations of inside and outside surface characteristics, internal properties of the wall, inside and outside air temperatures and intensity of incident radiation.

The present paper presents a practical method for calculating the heat flow into the interior space as a function of the thermal properties of the wall, the surface coefficients and the variable quantities of outdoor temperature and solar intensity when the wall structure is homogeneous. The solution to this problem

[1] Footnote references refer to numbers in bibliography.

was worked out by members of an advanced course in engineering.[4] Although the present paper gives no new experimental data, the accuracy of the method may be demonstrated by using data of Houghten et al [3] where the tests include not only measured rates of heat flow but also records of outdoor air temperature and pyroheliometer measurements of solar intensity.

ASSUMPTIONS

The basic assumptions employed in this analysis are:

1. The wall is of infinite length and height but of finite thickness. The assumption involves uni-directional heat flow, and is sufficiently accurate for any practical case.
2. The wall is homogeneous, which is sufficiently accurate for un-insulated masonry and brick construction.
3. Constant indoor and outdoor surface heat transfer coefficients prevail.
4. The thermal properties of conductivity and specific heat of the wall structure, and the density, are independent of temperature variations.
5. The variations of outdoor air temperature and solar intensity with time are identical on successive days.

Certain incidental assumptions are also employed to further simplify the statement of the method and the explanations of the analysis; these are:

1. The air inside the room is maintained at a constant temperature. (An assumption of periodic variation of inside temperature will not change the method.)
2. The absorptivity of the outside surface to solar radiation is independent of the angle of incidence (any assumption of variation of absorptivity with angle of incidence may be made without changing the method).

OUTSIDE SURFACE PHENOMENA

An outside wall surface exposed to the sun is subject to heating or cooling in the following ways:

1. By convection to or from the surrounding atmosphere.
2. By radiation from the sun.
3. By radiation to or from the sky or surrounding objects.

An illustration of the foregoing may be obtained by referring to Fig. 1. Radiant energy from the sun falling on the outside surface of the wall is partially reflected and partially absorbed. The reflected portion has no influence on the heat flow through the wall. The absorbed portion of the solar energy, however, results in a rise in the outside surface temperature of the wall, as noted previously. By virtue of this rise in temperature of the outside surface the temperature gradient through the wall is increased, resulting in a higher rate of heat flow into the room. Heat also flows in the opposite direction from the outside surface by convection to the atmosphere and by radiation to the sky and surrounding objects.

The intensity of the incident solar radiation per square foot of wall area depends on the orientation of the wall, the time of day, the time of year, the

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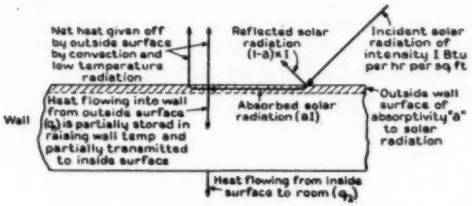
Net heat given off by outside surface by convection and low temperature radiation
Reflected solar radiation (1-a)×I
Incident solar radiation of intensity I Btu per hr per sq ft
Outside wall surface of absorptivity "a" to solar radiation
Wall
Heat flowing into wall from outside surface (q_1) is partially stored in raising wall temp and partially transmitted to inside surface
Absorbed solar radiation (aI)
Heat flowing from inside surface to room (q_2)

FIG. 1. HEAT FLOW PHENOMENA IN WALL EXPOSED TO THE SUN

latitude and the degree of cloudiness, smokiness, etc. Fig. 2, which is drawn from data presented previously,[1] illustrates the manner in which the incident solar radiation varies with orientation and time of day. The fraction of the incident radiation which is absorbed by the wall surface (absorptivity of the surface to solar radiation) depends upon the nature and color of the surface, and may also depend upon the angle of incidence. Thus, the solar heat input to one square foot of the wall can be expressed simply as aI, where a is the absorptivity of the surface to solar radiation and I is the instantaneous value of the sun intensity on one square foot of the wall surface.

In addition to receiving radiation from the sun, a square foot of wall surface emits to surrounding solid objects a net quantity of low temperature radiation, which in Btu per hour is given by

$$R - R' = 0.174 \times 10^{-8} \xi'' (T_1^4 - T_s^4) \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where

ξ'' = emissivity of low temperature radiation for the particular geometric relation between the radiating and receiving surface.

T_1 = temperature of the wall surface, degrees Fahrenheit, absolute.

T_s = temperature of surrounding solid objects, degrees Fahrenheit, absolute.

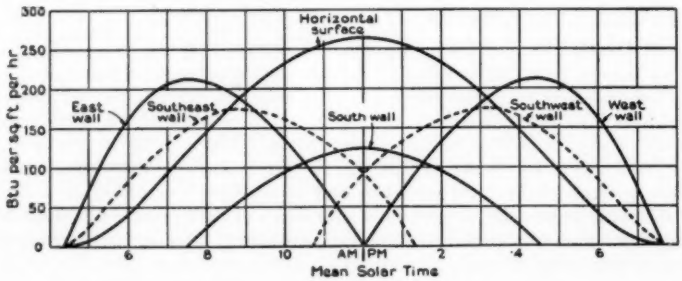

FIG. 2. AVERAGE MIDSUMMER INTENSITY OF SOLAR RADIATION ON VARIOUS WALL SURFACES AT 45 DEG N LATITUDE

Over the small temperature ranges encountered in the present study, $(R-R')$ can be approximated by

$$R - R' = \xi'' F (t'_1 - t_a) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

$$R - R' = h_r (t'_1 - t_a) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

where t'_1 and t_a are temperatures of the wall surface and surrounding solid objects in degrees Fahrenheit, and h_r is a radiation heat transfer coefficient equal to the product of ξ'' and F. F is a factor dependent upon the temperature level of the radiation heat transfer process, and is not the same as the F defined in a previous publication.[1] The possibility of expressing heat transfer by radiation in this manner when the temperature range is small has been pointed out by King.[5]

Horizontal surfaces, such as roofs, radiate to the sky an amount of heat, in Btu per hour per square foot,

$$R = 0.174 \times 10^{-8} \xi' T_1^4$$

where ξ' is the emissivity of the outside surface to low temperature radiation. However, the surface also receives an appreciable amount of radiant energy by radiation from water vapor in the lower regions of the atmosphere. For example, on a day when the air temperature at the surface is 70 F and a relative humidity of about 50 per cent prevails, approximately 105 Btu per hour per square foot is received by radiation from water vapor in the air (Appendix II). Since a black body at 70 F emits 137 Btu per hour per square foot, the net heat loss from a black horizontal surface at 70 F is approximately 32 Btu per hour per square foot.

For the sake of simplicity, the heat radiated and the heat absorbed by a surface facing the sky may be approximated, respectively, by

$$R = \xi' [R_b + F (t'_1 - t_b)] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4a)$$

$$R' = \xi' [R_b' + F' (t_a - t_b)] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4b)$$

where

R_b = heat radiated by a black body at a reference temperature t_b, Btu per hour per square foot.

R_b' = black-body absorption of radiation from an atmosphere whose temperature at the surface is t_b, Btu per hour per square foot.

t_a = temperature of surrounding air, degrees Fahrenheit.

F, F' = constants depending on the temperature level (F' is approximately 0.8 F.)

Convection heat transfer to a unit area of the outside surface is given by the familiar expression

$$h_1 (t_a - t'_1) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

where h_1 is the surface coefficient, dependent upon the wind velocity, and t_a and t'_1 are the instantaneous values of air temperature and outside surface temperature, respectively.

Air temperature as a function of the time of day varies so much from place to place that it is not possible to give a universal curve. It can be said, however, that it is nearly always a close approximation to an average value equal to the mean daily temperature plus an oscillating component with an amplitude

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equal to one half the *mean daily range*.[a] For most practical applications this can be approximated by a constant with a superposed sine wave having a 24-hour period. For more precise representation of the variation additional harmonics may be added to the expression by the method given in Appendix III. The maximum daily temperature usually occurs from one to three hours after noon, mean solar time (see Fig. 3).

The components of heat transfer at the outside surface have now all been examined and they may be combined to give the equation for the heat balance at the outside surface, as follows:

$$q_1 = h_1 (t_a - t'_1) + aI - (R - R') \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

where

q_1 = net heat flow from the outer surface to the interior of the wall, Btu per hour per square foot.

In most cases the assumption may be made, with sufficient accuracy, that the temperature of surrounding solid objects is equal to the air temperature.

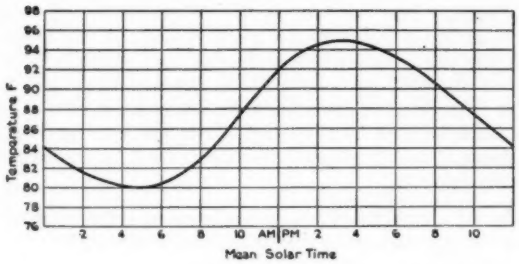

FIG. 3. APPROXIMATION OF TYPICAL VARIATION IN OUTDOOR AIR TEMPERATURE ON A DESIGN DAY

Hence, setting t_a equal to t_a, and substituting (2) in (6), the expression for walls facing terrestrial objects reduces to (7b), in which the radiation coefficient is lumped with the convection coefficient.

$$q_1 = (h_1 + \xi'' F) (t_a - t_1') + aI \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7a)$$
$$q_1 = H_1 (t_a - t'_1) + aI \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7b)$$

For surfaces facing the sky the simplifying assumption is made that F and F' in equations (4) are equal. Substituting in (6), the expression for surfaces facing the sky becomes

$$q_1 = (h_1 + \xi' F) (t_a - t'_1) + aI - \xi' (R_b - R_b') \quad . \quad . \quad . \quad . \quad (8a)$$
$$q_1 = H_1' (t_a - t'_1) + (aI - \xi' \Delta R_b) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8b)$$

Here a part of the low-temperature radiation effect is included in the outside film coefficient and the remainder is taken care of by the constant term $(\xi' \Delta R_b)$.

[a] The *mean daily range* is the mean maximum daily temperature minus the mean minimum daily temperature.

Both equations (7) and (8) contain a term which is dependent on temperature difference and another term which is independent of temperature difference but dependent on intensity of radiant energy. Each one of these terms may vary in magnitude from hour to hour, as explained previously. Hence, a single solution to the differential equation written for the flow of heat through the wall is applicable to surfaces facing either the sky or terrestrial objects. Most practical applications may be reduced to either one or the other of these two cases. However, whenever the problem warrants more detailed study it is possible to use a mean somewhere between the two limiting cases given.

CONDITIONS WITHIN THE WALL

An elementary thickness of wall just beneath the outside surface receives the net heat given to the wall by convection and radiation. A portion of this energy goes to heat up the element in question while the balance is conducted on to the next elementary thickness of the wall. Due to the fact that a certain time is required for the incident heat to raise the temperature of the element in question and pass on the remainder to the next element, there is a time lag between the input of a heat impulse at the outer surface and its arrival at the inner surface. Furthermore, because the incident heat impulse has had some of its energy used in heating up the wall, the amplitude of the corresponding impulse is considerably reduced by the time it reaches the inner surface.

Thus, if the radiation falling on the outside surface of a wall were sinusoidal about a zero average value, a sinusoidal heat input to the room would be found from this source, lower in amplitude than that of the incident radiation and somewhat later in time phase. Similarly, if the outdoor temperature varied sinusoidally about the same temperature as that of the indoors, a sinusoidal heat flow at the inner surface would be found lower in amplitude and later in phase than that induced across the outside surface by the variation in air temperature. Actually, of course, the net radiation on the outside surface is not zero but some positive value; likewise, the mean outdoor temperature differs from that maintained indoors by some constant amount. These constants give rise to a constant heat flow through the wall which, superimposed upon the oscillating components at the inner surface, gives the total heat flow into the room.

EXPRESSION FOR HEAT FLOW

Having gained a physical picture of what occurs in the wall, a brief outline of the method of solution for the heat flow into the room will now be given. All temperatures are measured with respect to an assumed constant indoor air temperature for simplicity of mathematical expression. No generality is lost by this assumption, however, and the final solution is expressed in terms of the difference between indoor and outdoor temperatures.

The differential equation holding at all points in the wall is

$$k \frac{\partial^2 t}{\partial x^2} = \rho c \frac{\partial t}{\partial \theta} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

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which states that the difference between the heat flow in at one side of an elementary thickness ∂x and that out at the other side causes a local rate of rise in temperature dependent upon the heat capacity, ρc.

This must be solved subject to the following boundary condition at the outside surface, which is simply a restatement of equation (8b):

$$-k\left(\frac{\partial t}{\partial x}\right)_1 = H_1'\,(t_s - t'_1) + aI - \xi'\Delta R_b \quad . \quad . \quad . \quad . \quad . \quad (10)$$

The term on the left side, being the product of the gradient just inside the outer surface times the conductivity, represents the heat input to the wall.

The boundary condition at the inside surface is

$$-k\left(\frac{\partial t}{\partial x}\right)_2 = h_2\,t_2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

which states that the heat flow through the innermost element of the wall equals that transferred by convection (and radiation) to the room.

The solution of the differential equation (9), subject to boundary conditions (10) and (11), is:

$$q_2 = Ut_m + [V_0t_0\cos(\omega_0\theta - \alpha_0 - \Phi_0) + V_1t_1\cos(\omega_1\theta - \alpha_1 - \Phi_1) \ldots\ldots]$$

$$+ \frac{U}{H'_1}\,(aI_m - \xi'\Delta R_b) + \frac{a}{H'_1}\,[V_0I_0\cos(\omega_0\theta - \beta_0 - \Phi_0) + V_1I_1\cos$$
$$(\omega_1\theta - \beta_1 - \Phi_1) + \ldots] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

This solution is derived in Appendix I and is based on the assumption that the temperature difference from outdoors to indoors and the intensity of incident solar heat can each be expressed as a constant plus a series of sinusoidal harmonics, as follows:

$$t = t_m + t_0\cos(\omega_0\theta - \alpha_0) + t_1\cos(\omega_1\theta - \alpha_1) + \quad . \quad . \quad . \quad . \quad . \quad (13)$$
$$I = I_m + I_0\cos(\omega_0\theta - \beta_0) + I_1\cos(\omega_1\theta - \beta_1) + \quad . \quad . \quad . \quad . \quad . \quad (14)$$

The symbols used in the expressions are defined at the end of the paper.

In equation (12) the term Ut_m represents the steady heat input to the room resulting from the excess of mean outdoor temperature over indoor temperature. This is the same as the well known expression for steady-state heat transfer by temperature difference used in all present standard methods of heat loss and heat gain calculations excepting that the temperature difference, t_m, is the mean temperature difference over a 24-hour period rather than the instantaneous difference between outdoor and indoor temperatures.

The term $\frac{U}{H_i'}\,(aI_m - \xi'\Delta R_b)$ is the heat input from mean solar radiation and net radiation to the sky. This term is equivalent to the expression for additional conduction through walls exposed to the sun given in a previous publication [1] excepting that the radiation intensity I_m is the average intensity over a 24-hour period rather than the instantaneous intensity, and excepting for the inclusion of a correction for net radiation to the sky.

Drawing an analogy with the electrical circuit, the two terms given may be likened to the direct current flowing when a constant voltage is applied, that is, $(1/U)$ is equivalent to the resistance of the circuit to direct current.

The first bracketed term in equation (12) is the oscillating component resulting from variation of outdoor temperature and the second bracketed term is the oscillating component resulting from variation in solar intensity. Referring again to the electrical analogy, these terms correspond with the current flowing in an electrical circuit having both a resistance and reactance to alternating voltages. That is, $(1/V_0, 1/V_1$, etc.) correspond to the impedances of the electrical circuit for alternating currents of various frequencies.

Drawing the electrical analogy still further, the angles Φ_0, Φ_1, etc. are equivalent to the phase displacement between the applied voltage and the resultant current.

The terms V_0, V_1, etc. and Φ_0, Φ_1, etc. in equation (12) are functions of the wall properties and the particular harmonic of air temperature or solar intensity being considered. The mathematical expressions for these terms, which are rather complicated, are given in Appendix I. However, the expressions for V and Φ can be plotted against simple functions of the wall properties, as explained, thereby avoiding complicated numerical work in the application of the method.

In order to visualize the effect of the heat storage capacity of the wall upon the magnitude of the transmitted heat impulse, V_0, V_1, etc. may be divided by U. Thus, if

$$\lambda = \frac{V}{U} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (15)$$

equation (12) may be re-written as follows:

$$\begin{aligned} q_2 = {} & U t_m + U [\lambda_0 I_0 \cos(\omega_0\theta - \alpha_0 - \Phi_0) + \lambda_1 I_1 \cos(\omega_1\theta - \alpha_1 - \Phi_1) + \dots .] \\ & + \frac{U}{H_1'} (aI_m - \xi' \Delta R_b) + \frac{U}{H_1'} a [\lambda_0 I_0 \cos(\omega_0\theta - \beta_0 - \Phi_0) \\ & + \lambda_1 I_1 \cos(\omega_1\theta - \beta_1 - \Phi_1) + \dots] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (16) \end{aligned}$$

The solution given has been written for the case of a horizontal surface facing the sky, which gives the more general expression for the constant term in the radiation component. However, the solution for vertical walls facing terrestrial objects is identical, excepting that the term

$$\frac{U}{H_1'} (aI_m - \xi' \Delta R_b) \text{ becomes } \dots \frac{U}{H_1} (aI_m)$$

Furthermore, for practical purposes it is reasonable to make the assumption that H'_1 and H_1 are equal for the following reasons:

1. For low temperature radiation the emissivity of the outer surface of most wall constructions, of surrounding solid objects and of the sky may be assumed nearly equal to unity. This makes it possible to neglect the configuration factor in determining the effective emissivity of the outer wall surface.
2. The radiation component, h_r, is from 25 to 50 per cent of the outside film coefficient H_1. Hence, any error in the combined film coefficient is not over one-half the error in the assumed radiation coefficient. and is usually less.

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3. The percentage error in the calculated heat flow is less than the percentage error in the assumed film coefficient.

The factor λ in (16) is the ratio of the rate of heat transmission through the wall for a sinusoidally variable temperature difference to the rate of heat transmission that would exist if the same temperature difference were maintained constant for an indefinite period of time. It is, therefore, a decrement factor. It is instructive to note that the same decrement factor λ and the same lag angle Φ occur in both the sun effect term and in the term resulting from outdoor temperature variation. In other words, both the decrement factor and the lag angle are functions of the wall properties and are independent of the means by which the heat is received.

Under certain circumstances, harmonics of the temperature and solar intensity waves may be neglected. Not only are the amplitudes of the impulses smaller than that of the fundamental, but the effect felt at the inner surface is

TABLE 1—VALUES OF ω AND RELATIVE VALUES OF σk FOR DIFFERENT SINUSOIDAL HARMONICS

HARMONIC	FREQUENCY, CYCLES PER DAY	PERIOD, HOURS PER CYCLE	ω	RELATIVE VALUE OF (σk)
Fundamental	1	24	0.2618	1.00
1st harmonic	2	12	0.5236	1.41
2nd harmonic	3	8	0.7854	1.73
3rd harmonic	4	6	1.0472	2.00

smaller because of a lower value of λ for the harmonics. This will be demonstrated later. In any case, it is very seldom that harmonics higher than the first must be considered.

Figs. 4 and 5 give charts of λ and Φ vs. the wall characteristic and the frequency of the variation in temperature or solar intensity. The variables against which λ and Φ are plotted in Figs. 4 and 5 are σk and L/k, where

$$\sigma = \sqrt{\frac{\rho c \omega}{2k}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (17)$$

ρ = density of wall structure, pounds per cubic foot.
c = specific heat of wall structure, Btu per pound per degree Fahrenheit.
k = conductivity of wall structure, Btu per hour per square foot per degree Fahrenheit per foot.
L = thickness of wall, feet.
ω = angular velocity of sinusoidal variation, radians per hour.

It can be seen that L/k is constant for any given wall, but that σk depends on whether the fundamental or one of the harmonics of the variation is being considered. Table 1 gives values of ω and the relative magnitudes of σk for several harmonics.

The use of Figs. 4 and 5 will now be explained by means of an example. In order to preserve simplicity, it will be assumed that the wall is subjected

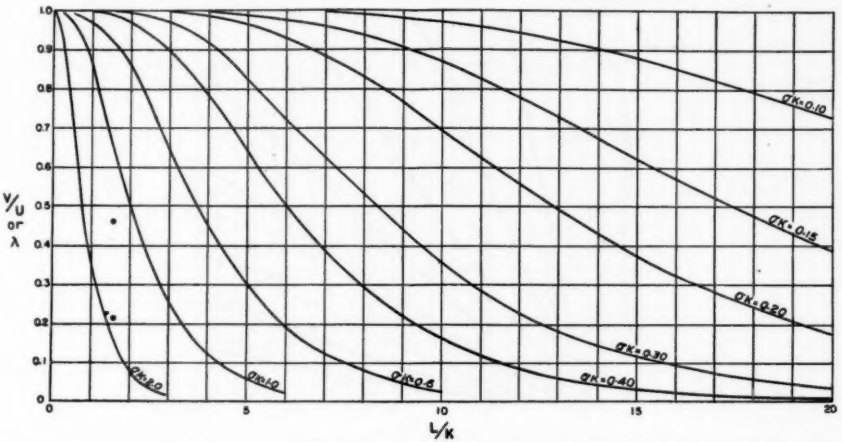

FIG. 4. DECREMENT FACTORS FOR SINUSOIDAL COMPONENTS OF HEAT FLOW THROUGH HOMOGENEOUS WALLS

to a variation in temperature, but that it is not exposed to the sun. The assumed variation in temperature difference between indoors and outdoors is

$$t = 10 + 15 \cos (15\,\theta - 210) + 2 \cos (30\,\theta - 240)$$

This corresponds to a maximum temperature difference of 23 F at 2 P. M. The assumed wall characteristics are:

$$L/k = 7.0$$
$$\sigma k = 0.3 \text{ (for the fundamental)}$$
$$U = 0.128$$

Referring to Table 1, σk for the first harmonic can be determined, and then from Figs. 4 and 5 λ and Φ can be found for the fundamental and the first harmonic. Table 2 summarizes the numerical results and also shows the magnitude of the mean, the fundamental and the first harmonic of the heat flow through the inside wall.

Neglecting the contribution of the first harmonic to the total heat flow (resulting in an error of not more than 3 per cent), the maximum flow of heat through the inside surface of the wall is 2.47 Btu per hour per square foot. With the same over-all coefficient of heat transfer in a wall with zero heat storage capacity and with the same temperature difference, the maximum flow of heat would be

$$0.128 \times 23 = 2.94 \text{ Btu per hour.}$$

To summarize, the effect of heat storage capacity in this example is to reduce the maximum rate of heat flow by 16 per cent and to delay the time when the maximum flow across the interior surface occurs by approximately 5.8 hours.

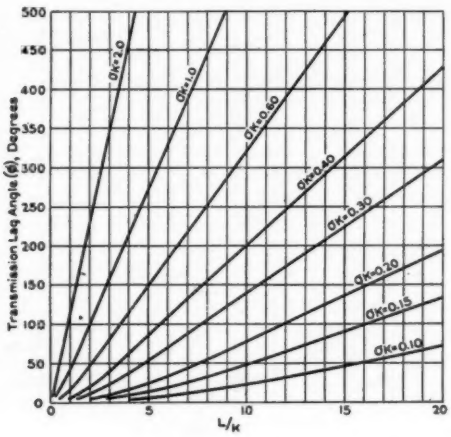

FIG. 5. TRANSMISSION LAG ANGLES FOR SINUSOIDAL COMPONENTS OF HEAT FLOW THROUGH HOMOGENEOUS WALLS

CORRELATION WITH TEST DATA

An interesting check on the method which has been presented is obtained by comparing calculated heat flows at the inner surface with those obtained in tests by Houghten et al.[3] The following examples are for two different panels, one tested on July 27, 1931 and the other on September 10, 1931. These are the only two days on which the reported test data include measurements of both outdoor air temperature and intensity of solar radiation.

The first case is that of a 2-in. thick horizontal pine panel tested on July 27, 1931. The solar intensity on a horizontal surface for that day is shown in

TABLE 2—SUMMARY OF ILLUSTRATIVE EXAMPLE

COMPONENT	L/k	σk	λ (FIG. 4)	I (MAX. VALUE)	$q = \lambda U I$	ϕ (FIG. 5)	
						Deg.	Hours
Mean	7.0	..	..	10	1.28	..	..
Fundamental	7.0	0.30	0.62	15	1.19	87	5.8
1st Harmonic	7.0	0.42	0.33	2	0.08	140	4.7

Fig. 6a. When analyzed by the method given in Appendix III, this may be expressed as

$$I = 58 - 70.5\ cos\ 15\theta - 8.3\ sin\ 15\theta + 38.5\ cos\ 30\theta + 3.3\ sin\ 30\theta + \ldots$$
$$= 58 + 71.0\ cos\ (15\theta - 187) + 38.6\ cos\ (30\theta - 5) + \ldots \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (18)$$

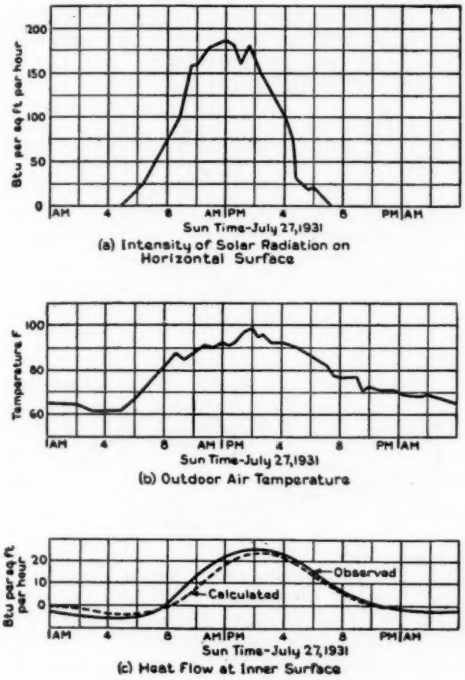

FIG. 6. TEST ON 2-IN. PINE PANEL, JULY 27, 1931

(a) Solar intensity on horizontal surface
(b) Outdoor air temperature
(c) Heat flow into room

The outdoor air temperature for the same day is shown in Fig. 6b. In a similar manner it is expressed as

$$t = 9.4 - 13.9\ cos\ 15\theta - 8.8\ sin\ 15\theta + 1.7\ cos\ 30\theta - 1.0\ sin\ 30\theta + \ldots$$
$$= 9.4 + 16.4\ cos\ (15\theta - 212) + 2.0\ cos\ (30\theta - 329) + \ldots \quad . \quad . \quad . \quad . \quad . \quad . \quad (19)$$

where the indoor temperature of 69.6 F is the zero reference.

Using the following numerical values given in the previous paper,[3] the heat flow at the inner surface was calculated through the use of equation (12) and the expressions for V and Φ are given in detail in Appendix I.

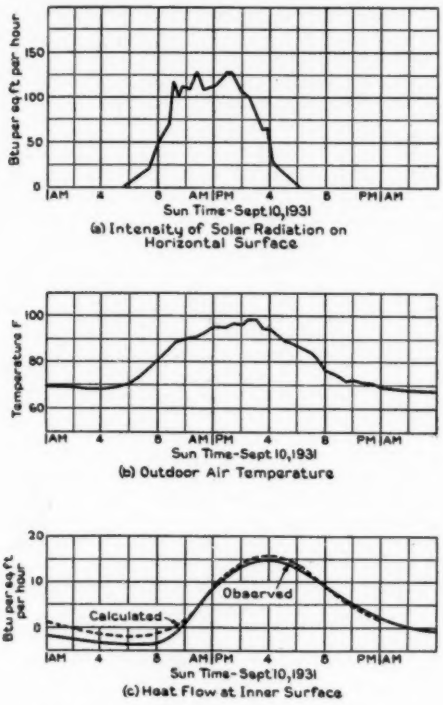

FIG. 7. TEST ON 4-IN. GYPSUM PANEL, SEPTEMBER 10, 1931

(a) Solar intensity on horizontal surface
(b) Outdoor air temperature
(c) Heat flow into room

H_1 = 2.0 Btu per hour − ft² − F
h_2 = 1.9 Btu per hour − ft² − F
k = 0.069 Btu per hour − ft − F
c = 0.467 Btu per pound − F
ρ = 37.5 lb per cubic foot
L = 0.180 ft
ΔR_b = 32 Btu per hour − ft² (supplied by authors)
$a = \xi'$ = 1.0 (supplied by authors)

The assumption of unity for absorptivity to solar radiation and emissivity to low temperature radiation is based on the explanation [3] that the outside surfaces of the panels were painted with lampblack pigment.

The calculated results are plotted as the broken line in Fig. 6c. The solid line in the same figure indicates the test results. The agreement is well within

limits with which atmospheric conditions could be predicted for design purposes.

The second case is that of a 4-in. thick gypsum panel, similarly oriented, and tested on September 10, 1931. The curve of solar intensity for that day is given in Fig. 7a, and may be mathematically expressed as

$$I = 37 - 59.6 \cos 15\theta + 2.3 \sin 15\theta + 27.0 \cos 30\theta - 1.9 \sin 30\theta + \ldots$$
$$= 37 + 59.6 \cos (15\theta - 178) + 27.4 \cos (30\theta - 356) + \ldots \quad (20)$$

Outdoor air temperature for the day is shown in Fig. 7b and may be expressed as

$$t = 10.4 - 13.3 \cos 15\theta - 7.4 \sin 15\theta + 1.6 \cos 30\theta + 1.1 \sin 30\theta + \ldots$$
$$= 10.4 + 15.2 \cos (15\theta - 209) + 2.0 \cos (30\theta - 35) + \ldots \quad (21)$$

where the indoor temperature of 69.6 F is the zero reference.

Constants used in calculating the heat flow at the inner surface as before are:

h_1 = 2.0 Btu per hour − ft² − F
h_2 = 1.9 Btu per hour − ft² − F
k = 0.12 Btu per hour − ft − F
c = 0.234 Btu per pound − F
ρ = 64.9 lb per ft³
L = 0.350 ft
R_b = 32 Btu per hour − ft² (supplied by authors)
$a = \xi'$ = 1.0 (supplied by authors)

The results are plotted as the broken line in Fig. 7c. Again a satisfactory agreement with test results is obtained.

SAMPLE CALCULATION

A complete numerical example is given to illustrate the application of the method which has been given to a homogeneous wall exposed to the sun.

Consider an 8-in. brick wall with an unobstructed southern exposure at latitude 45 deg. Assume that for design conditions the solar intensity as given by Fig. 2 may be used, and that the outdoor air temperature given for July 27, 1931, at Pittsburgh, Pa.[3] may be used. Assume, also, that the design indoor temperature is 80 F. The equations for solar intensity and temperature difference, outdoors-to-indoors, may be derived by the method given in Appendix III.

This results in the following, where all harmonics above the first are neglected:

$$I = 29.4 + 52.4 \cos (15\theta - 180) + 34.8 \cos 30\theta \quad (22)$$
$$t_o = 79.0 + 16.4 \cos (15\theta - 212) + 2.0 \cos (30\theta - 329) \quad (23)$$
$$t = -1 + 16.4 \cos (15\theta - 212) + 2.0 \cos (30\theta - 329) \quad (24)$$

where

t_o = outdoor air temperature
t = difference between outdoor and indoor air temperatures

The wall constants are now evaluated. The following quantities are obtained from the references indicated:

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$k = 0.42$ Btu per hour − ft − F [6]
$\rho = 132$ lb per ft³ [7]
$c = 0.22$ Btu per pound − F [7]
$h_2 = 1.65$ Btu per hour − ft² − F [8]
$H_1 = 4.3$ Btu per hour − ft² − F (supplied by authors)[b]
$a = 0.7$ [1]

For the fundamental wave, $\omega_0 = 2\pi/24 = 0.262$ radians per hour or 15 deg per hour. For the first harmonic, $\omega_1 = 2\pi/12 = 0.524$ radians per hour or 30 deg per hour. From the above,

$$\sigma_0 k = \sqrt{\frac{\rho c k \omega_0}{2}} = \sqrt{\frac{(132)\ (0.22)\ (0.42)\ (0.262)}{2}} = 1.27$$

and

$$\sigma_1 k = \sqrt{\frac{\rho c k \omega_1}{2}} = \sqrt{\frac{(132)\ (0.22)\ (0.42)\ (0.524)}{2}} = 1.80$$

$$\text{The factor } L/k = \frac{(8)}{(12)\ (0.42)} = 1.58$$

The over-all heat transfer coefficient of the wall is

$$U = \frac{1}{\frac{1}{H_1} + \frac{1}{h_2} + \frac{L}{k}} = \frac{1}{\frac{1}{4.3} + \frac{1}{1.65} + 1.58} = 0.41$$

Referring to Figs. 4 and 5, the following values of λ and Φ are found for the previous values of σk and L/k:

$\lambda_0 = 0.46$ $\qquad$ $\Phi_0 = 106$
$\lambda_1 = 0.21$ $\qquad$ $\Phi_1 = 163$

Finally, it is assumed that the wall faces either terrestrial objects of unit emissivity and atmospheric temperature, or layers of air near the earth of sufficient thickness to be effectively a black body. Here $\Delta R_b = 0$.

All constants required in equation (16) have now been determined. Substituting these constants, the equation for heat flow through one square foot of the inner surface of the wall in question is:

$$q_2 = (0.41)\ (-1.0) + (0.41)\left[\begin{array}{l}(0.46)\ (16.4)\ \cos\ (15\theta - 212 - 106) \\ +\ (0.21)\ (2.0)\ \cos\ (30\theta - 329 - 163)\end{array}\right]$$

---

[b] This numerical value, and Figs. 4 and 5 are based on surface coefficients, as follows:
$H_1 = 4.3$ (supplied by authors)
$h_2 = 1.65$ [8]

The outside surface coefficient is based on a wind velocity of 8.1 mph, which is the average summer velocity for 69 cities listed in the 1938 GUIDE.[9] It is derived from the standard value of 6.0[8] based on 15 mph wind velocity and used for winter design calculations, as follows:

From King,[5] the effective radiation coefficient for black bodies at ordinary temperature ranges is 1.3. From the same source, the convection coefficient varies as the 0.75 power of velocity. Hence

$$(H_1)\ 8.1\ \text{mph} = 1.3 + (6.0 - 1.3)\left(\frac{8.1}{15}\right)^{0.75} = 4.26$$

The assumption that outside surfaces of average wall structures are very nearly black bodies to low temperature radiation is sufficiently correct for practical purposes. The error involved is small, as an assumption of $e = 0.75$ results in a combined outside coefficient of 4.14 in place of 4.26

$$+ \frac{(0.41)}{(4.3)}(0.7)(29.4) + \frac{(0.41)}{(4.3)}(0.7)\left[\begin{matrix}(0.46)(52.4)\cos(15\theta - 180 - 106) \\ + (0.21)(34.8)\cos(30\theta - 163)\end{matrix}\right]$$

$$q_2 = 1.53 + 3.10\cos(15\theta - 318) + 0.17\cos(30\theta - 132) + 1.61\cos(15\theta - 286) + 0.49\cos(30\theta - 163) \quad (25)$$

Equation (25) may be plotted easily over whatever period is of interest. Fig. 8 gives it in its entirety. Fig. 8 also shows, for purposes of comparison, what the calculated heat flow would be if heat storage capacity were neglected. This is calculated from Equation (26) which is the same as (25), excepting that $\lambda_0 = \lambda_1 = 1.0$ and $\Phi_0 = \Phi_1 = 0$.

$$q_2 = 1.55 + 6.72\cos(15\theta - 212) + 0.82\cos(30\theta - 329) + 3.50\cos(15\theta - 180) + 2.32\cos 30\theta \quad (26)$$

CONCLUSIONS

1. This paper has presented a rational method for calculating the heat flow through a homogeneous wall that experiences a diurnal variation of air temperature and solar

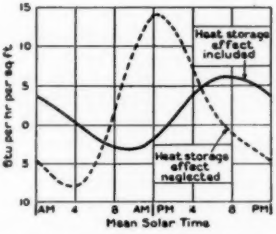

FIG. 8. EFFECT OF HEAT STORAGE ON HEAT TRANSMISSION THROUGH AN 8-IN. BRICK WALL

intensity on its outside surface, and is bordered at the inside by room air at constant temperature. The method avoids a determination or estimate of the outside surface temperature of the wall, and gives the heat flow into the room, as a function of time, in terms of the wall constants and the design conditions of solar intensity and outdoor air temperature. The analysis shows that a heat impulse of a given amplitude entering the wall at the outer surface requires an appreciable time to pass through, and that the amplitude of the heat impulse is reduced. The amount of time lag and the reduction in amplitude depend upon the thickness and thermal properties of the wall, and can be determined easily from charts.

2. It is believed that the method presented here is sufficiently simple so that it can be used in practical applications where certain design outdoor and indoor conditions are assumed. However, some further simplifications can undoubtedly be made to permit consideration of the influence of the heat storage effect with a minimum of time and effort. Tabulations of wall properties not easily found at present in standard handbooks and more extensive charts than those presented here will be of material benefit.

3. The method presented in this paper does not solve the problem for non-homogeneous walls, which is much more complicated than that for homogeneous walls. However, the authors are of the opinion that further analysis and further comparison

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with test results on various types of wall construction will permit the development of a single method which may be applied with practical accuracy to any type of wall structure; that further study will lead to the development of reasonably simple methods which can be used to determine constants for various wall structures to be used in such a method.

NOTATION

Letter symbols used in this paper are defined as follows:

a = absorptivity of outer wall surface to solar radiation.
c = specific heat of wall structure, Btu per pound − deg F.
F = effective radiation heat transfer coefficient for black body, Btu per hour − ft² − deg F.
F' = effective radiation heat transfer coefficient for radiation from the atmosphere, Btu per hour − ft² − deg F.
h = convection surface heat transfer coefficient, Btu per hour − ft² − deg F.
H = combined convection and radiation surface heat transfer coefficient, Btu per hour − ft² − deg F.
I = intensity of solar radiation on wall surface, Btu per hour − ft².
k = thermal conductivity of wall structure, Btu per hour − ft − deg F.
L = total wall thickness, feet.
η = heat flow, Btu per hour − ft².
R = outgoing low temperature radiation, from wall, Btu per hour − ft².
R' = low temperature radiation received by wall, Btu per hour − ft².
R_b = value of R at a reference temperature, Btu per hour − ft².
R_b' = value of R' at a reference temperature, Btu per hour − ft².
ΔR_b = $R_b - R_b'$.
t = temperature, degrees Fahrenheit, usually referred to indoor temperature, for zero.
T = temperature, degrees Fahrenheit, absolute.
U = over-all heat transfer coefficient of wall to constant temperature difference, Btu per hour − ft² − deg F.
V = over-all heat transfer coefficient of wall to sinusoidally varying temperature.
x = distance through the wall, feet, measured from the center.
α = displacement angle of outdoor air temperature wave referred to midnight as zero, radians.
β = displacement angle of solar intensity wave referred to midnight as zero, radians.
ξ = emissivity to low temperature radiation.
λ = decrement factor for amplitude of transmitted heat impulse.
λ = V/U.
ρ = density of wall structure, pounds per ft³.
ϕ = lag angle of transmitted heat impulse, radians.
ω = angular velocity of sinusoidal waves, radians per hour.

$$\sigma = \sqrt{\frac{\rho c \omega}{2k}}$$

θ = mean solar time, number of hours measured from midnight.

Subscript symbols are used as follows.

m refers to mean value of heat impulse.
o refers to fundamental component of sinusoidal variation (1 cycle per day).
1 refers to first harmonic component of sinusoidal variation (2 cycles per day).
1 refers to conditions at the outside surface of the wall in all other cases.
2 refers to second harmonic component of sinusoidal variation (3 cycles per day).
2 refers to conditions at the inside surface of the wall in all other cases.
s refers to surrounding terrestrial objects.

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APPENDIX I—MATHEMATICAL SOLUTION OF THE HEAT FLOW EQUATION

This Appendix gives the derivation for the heat flow through the inner surface of a homogeneous wall which is exposed on the outside to a variable air temperature and a variable intensity of solar radiation. The analysis employs the method of the complex variable.

In addition to the letter symbols given in the main body of the paper, the following are used in this Appendix:

$d = \frac{L}{2}$ = one half the wall thickness

e = base of natural logarithms

$m = \sqrt{j} = \sqrt[4]{-1}$

$j = \sqrt{-1}$

f, g, Y, Z = functions of wall properties

A, B, C, D, p = constants

A section of the wall is shown in Fig. I-1. For convenience, x is measured from the midpoint. The general differential equation for unidirectional heat flow in the x direction is

$$\frac{\partial^2 t}{\partial x^2} = \frac{\rho c}{k} \frac{\partial t}{\partial \theta} \quad \ldots\ldots\ldots\ldots \text{(I-1)}$$

For the sake of simplicity, only the constant term and one oscillating component term (the fundamental) of t and I will be used in the solution which follows. At the end, the solution for any harmonic may be written at once from the solution for the fundamental. This may then be superposed upon the basic solution to obtain the more complete and accurate one.

This must be solved subject to the following boundary conditions:

(1) At the inside surface ($x = d$)

$$-k\left(\frac{\partial t}{\partial x}\right)_2 = h_2 t_2 \quad \ldots\ldots\ldots\ldots \text{(I-2)}$$

where t_2 is the temperature of the inside wall surface referred to room air as zero.

(2) At the outside surface ($x = -d$)

$$-k\left(\frac{\partial t}{\partial x}\right)_1 = h_1 (t_a - t_a') + aI - R + R' \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(I-3)}$$

where t'_1 and t_a are temperatures of the outer wall surface and of outside air, respectively.

As an approximation, R and R' can be represented, respectively, by

$$R = \xi' [R_b + F (t_1' - t_b)] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(I-4a)}$$

$$R' = \xi' [R_b' + F' (t_a - t_b)] \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(I-4b)}$$

where R_b and R'_b are values of low temperature radiation from the outer wall surface and from the atmosphere, respectively, at a reference temperature t_b.

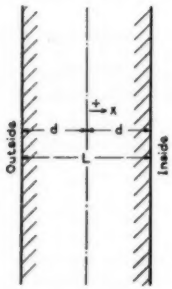

FIG. I-1. CROSS SECTION OF WALL

In Appendix II it is shown that $F' \cong 0.8$ F, so that it is fair to assume, at least for values of t_1' and t_a not too far removed from t_b in absolute temperature, that $F = F'$.

Then (I-3) becomes

$$\begin{aligned} -k\left(\frac{\partial t}{\partial x}\right)_1 &= (h_1 + \xi' F) (t_a - t_1') + aI - \xi' (R_b - R_b') \\ &= H_1 (t_a - t_1') + aI - \xi' (R_b - R_b') \\ &= H_1 \{ [t_m + t_0 \cos (\omega_0\theta - \alpha_0) + t_1 \cos (\omega_1\theta - \alpha_1) + \ldots] - t_1' \} \\ &\quad + a \{ I_m + I_0 \cos (\omega_0\theta - \beta_0) + I_1 \cos (\omega_1\theta - \beta_1) \ldots \} - \xi' \Delta R_b \quad . \quad . \quad \text{(I-3a)} \end{aligned}$$

where t_m, t_0, t_1, etc. refer to outdoor air temperature and t_1' refers to the temperature of the outer wall surface, both referred to inside or room air as zero.

Assume a solution to (I-1) of the form

$$t = A + Bx + e^{-m^2\omega_0\theta} (C \cos p_0 mx + D \sin p_0 mx) \quad . \quad . \quad . \quad . \quad \text{(I-5)}$$

in which A, B, C and D may be either real or complex numbers. Let it be understood that the real part of the resulting complex value of t will constitute the actual physical solution.

Substituting (I-5) into (I-1), there results

$$e^{-m^2\omega_0\theta} (-Cp_0^2m^2 \cos p_0 mx - Dp_0^2m^2 \sin p_0 mx)$$

$$= -\frac{\rho c m^2 \omega_0}{k} e^{-m^2\omega_0\theta} (C \cos p_0 mx + D \sin p_0 mx) \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(I-6)}$$

Since this must hold for all values of θ and x, it is an identity and the coefficients of corresponding terms can be equated to obtain

$$p_0 = \sqrt{\frac{pc\omega_0}{k}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(I-7)}$$

With this value for p_0 determined, (I-5) has been made to satisfy the differential equation (I-1).

In order to evaluate A, B, C and D, (I-5) is substituted into the boundary condition equations (I-2) and (I-3), reduced to complex form. Substituting (I-5) into the complex equivalent of (I-3a)

$$-B - e^{-m^2\omega_0\theta}\,[Cp_0m\ sin\ p_0md + Dp_0m\ cos\ p_0md]$$

$$= \frac{H_1t_m + aI_m - \xi'\Delta R_b}{k} + e^{-m^2\omega_0\beta}\left[\frac{(H_1t_0e^{m^2\alpha_0} + aI_0e^{m^2\beta_0})}{k}\right]$$

$$-\frac{H_1}{k}\left[A - Bd + e^{-m^2\omega_0\beta}\,(C\ cos\ p_0md - D\ sin\ p_0md)\right] \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(I-8)}$$

Substituting (I-5) into (I-2)

$$-B - e^{-m^2\omega_0\theta}\,[-Cp_0m\ sin\ p_0md + Dp_0m\ cos\ p_0md]$$

$$= \frac{h_2}{k}\left[A + Bd + e^{-m^2\omega_0\theta}\,(C\ cos\ p_0md + D\ sin\ p_0md)\right] \quad . \quad . \quad . \quad . \quad \text{(I-9)}$$

The boundary conditions must, of course, hold true at all values of θ; hence (I-8) and (I-9) are identities in which the coefficients of the $e^{-m^2\omega_0\theta}$ terms on either side may be equated. For the same reason, the constants on either side may be equated. Writing equations of the constants and coefficients of (I-8) respectively.

$$-B = \frac{H_1t_m + aI_m - \xi'\Delta R_b}{k} - \frac{H_1}{k}\,(A - Bd) \quad . \quad . \quad . \quad \text{f (I-10)}$$

and

$$Cp_0m\ sin\ p_0md + Dp_0m\ cos\ p_0md = -\frac{H_1}{k}\,(C\ cos\ p_0md - D\ sin\ p_0md)$$

$$+ \frac{H_1t_0e^{m^2\alpha_0} + aI_0e^{m^2\beta_0}}{k} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(I-11)}$$

Similarly, the equations of the constants and coefficients of (I-9) are, respectively,

$$-B = \frac{h_2}{k}\,(A + Bd) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(I-12)}$$

$$Cp_0m\ sin\ p_0md - Dp_0m\ cos\ p_0md = \frac{h_2}{k}\,(C\ cos\ p_0md + D\ sin\ p_0md) \quad . \quad \text{(I-13)}$$

Solving (I-10) and (I-12) simultaneously,

$$A = \frac{(k + dh_2)\,(H_1t_m + aI_m - \xi'\Delta R_b)}{k(H_1 + h_2) + 2dH_1h_2} \quad . \quad . \quad . \quad . \quad \text{(I-14)}$$

and

$$B = -\frac{h_2(H_1t_m + aI_m - \xi'\Delta R_b)}{k(H_1 + h_2) + 2dH_1h_2} \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(I-15)}$$

Solving (I-11) and (I-13) simultaneously while making use of the fact that

$$m = \frac{1 + m^2}{\sqrt{2}},\ sin\ m^2pd = m^2\ sinh\ pd \text{ and } cos\ m^2pd = cosh\ pd,$$

$$C = \frac{H_1t_0e^{m^2\alpha_0} + aI_0e^{m^2\beta_0}}{k(f_0 + m^2g_0)}\,(p_0m\ cos\ p_0md + \frac{h_2}{k}\ sin\ p_0md) \quad . \quad . \quad . \quad . \quad \text{(I-16)}$$

and

$$D = \frac{H_1 t_0 e^{m^2 a_0} + a I_0 e^{m^2 \beta_0}}{k(f_0 + m^2 g_0)} \left(p_0 m \cos p_0 m d + \frac{h_2}{k} \sin p_0 m d\right) \quad . \quad . \quad . \quad . \quad . \quad \text{(I-17)}$$

where

$$f_0 = \frac{H_1 h_2}{k^2} \sin \sqrt{2}\, p_0 d \cosh \sqrt{2}\, p_0 d + p_0^2 \cos \sqrt{2}\, p_0 d \sinh \sqrt{2}\, p_0 d$$

$$+ \frac{p_0(H_1 + h_2)}{\sqrt{2}\, k} (\cos \sqrt{2}\, p_0 d \cosh \sqrt{2}\, p_0 d + \sin \sqrt{2}\, p_0 d \sinh \sqrt{2}\, p_0 d) \; . \quad \text{(I-18)}$$

$$g_0 = \frac{H_1 h_2}{k^2} \cos \sqrt{2}\, p_0 d \sinh \sqrt{2}\, p_0 d - p_0^2 \sin \sqrt{2}\, p_0 d \cosh \sqrt{2}\, p_0 d$$

$$+ \frac{p_0(H_1 + h_2)}{\sqrt{2}\, k} (\cos \sqrt{2}\, p_0 d \cosh \sqrt{2}\, p_0 d - \sin \sqrt{2}\, p_0 d \sinh \sqrt{2}\, p_0 d). \quad \text{(I-19)}$$

With the constants of (I-5) thus evaluated so as to satisfy the boundary conditions, it is possible to find t_2 which, when multiplied by h_2, gives the heat input to the room. Substituting (I-14), (I-15), (I-16) and (I-17) into (I-5) and evaluating it at $x = d$, after simplifying and rationalizing,

$$t_2 = \frac{H_1 t_m + a I_m - \xi' \Delta R_b}{H_1 + h_2 + \dfrac{H_1 h_2 L}{k}}$$

$$+ \frac{H_1 t_0 p}{k(f_0^2 + g_0^2)} \left\{ \begin{array}{l} \left[\dfrac{f_0 + g_0}{\sqrt{2}} \cos(\omega_0\theta - \alpha_0) + \dfrac{f_0 - g_0}{\sqrt{2}} \sin(\omega_0\theta - \alpha_0) \right] \\ + \mathrm{j}\left[\dfrac{f_0 - g_0}{\sqrt{2}} \cos(\omega_0\theta - \alpha_0) - \dfrac{f_0 + g_0}{\sqrt{2}} \sin(\omega_0\theta - \alpha_0) \right] \end{array} \right\}$$

$$+ \frac{a I_0 p}{k(f_0^2 + g_0^2)} \left\{ \left[\frac{f_0 + g_0}{\sqrt{2}} \cos(\omega_0\theta - \beta_0) + \frac{f_0 - g_0}{\sqrt{2}} \sin(\omega_0\theta - \beta_0) \right] \right.$$

$$\left. + j\left[\frac{f_0 - g_0}{\sqrt{2}} \cos(\omega_0\theta - \beta_0) - \frac{f_0 + g_0}{\sqrt{2}} \sin(\omega_0\theta - \beta_0) \right] \right\} \quad . \quad . \quad . \quad . \quad \text{(I-20)}$$

Since it was understood at the beginning that the real part of t would constitute the physical solution, the imaginary part of (I-20) can be discarded. Multiplying the remainder by h_2, the heat input to the room is

$$h_2 t_2 = \frac{h_2(H_1 t_m + a I_m - \xi' \Delta R_b)}{H_1 + h_2 + \dfrac{H_1 h_2 L}{k}}$$

$$+ \frac{h_2 H_1 t_0 p}{k(f_0^2 + g_0^2)} \left[\frac{f_0 + g_0}{\sqrt{2}} \cos(\omega_0\theta - \alpha_0) + \frac{f_0 - g_0}{\sqrt{2}} \sin(\omega_0\theta - \alpha_0) \right]$$

$$+ \frac{h_2 a I_0 p}{k(f_0^2 + g_0^2)} \left[\frac{f_0 + g_0}{\sqrt{2}} \cos(\omega_0\theta - \beta_0) + \frac{f_0 - g_0}{\sqrt{2}} \sin(\omega_0\theta - \beta_0) \right] \quad . \quad \text{(I-21)}$$

To make the numerical calculation easier, let

$$\sigma_0 = \frac{p}{\sqrt{2}} = \sqrt{\frac{\rho c \omega_0}{2k}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(I-22)}$$

$$Y_0 = \frac{f_0 + g_0}{2\sigma_0^3} = \left(\frac{H_1 h_2}{2\sigma_0^2 k^2} + 1\right) \cos \sigma_0 L \sinh \sigma_0 L + \left(\frac{H_1 h_2}{2\sigma_0^2 k^2} - 1\right) \sin \sigma_0 L \cosh \sigma_0 L$$

$$+ \left(\frac{H_1 + h_2}{\sigma_0 k}\right) \cos \sigma_0 L \cosh \sigma_0 L \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(I-23)}$$

$$Z_0 = \left(\frac{H_1 h_2}{2\sigma_0^2 k^2} + 1\right) \sin \sigma_0 L \cosh \sigma_0 L - \left(\frac{H_1 h_2}{2\sigma_0^2 k^2} - 1\right) \cos \sigma_0 L \sinh \sigma_0 L$$

$$+ \left(\frac{H_1 + h_2}{\sigma_0 k}\right) \sin \sigma_0 L \sinh \sigma_0 L \quad \ldots \quad \text{(I-24)}$$

Finally, define an angle Φ such that

$$\sin \Phi_0 = \frac{f_0 - g_0}{\sqrt{2}\sqrt{f_0^2 + g_0^2}}$$

$$\cos \Phi_0 = \frac{f_0 + g_0}{\sqrt{2}\sqrt{f_0^2 + g_0^2}} \quad \ldots \quad \text{(I-25)}$$

from which

$$\Phi_0 = \tan^{-1} \frac{Z_0}{Y_0} \quad \ldots \quad \text{(I-26)}$$

where $\sin \Phi_0$ has the sign of Z_0
$\cos \Phi_0$ has the sign of Y_0

Simplifying (I-21) with the aid of (I-23), (I-24), and (I-25), the solution for the constant term and the fundamental is:

$$q_2 = h_2 t_2 = \frac{h_2(H_1 t_m + a I_m - \xi' \Delta R_b)}{H_1 + h_2 + \frac{H_1 h_2 L}{k}}$$

$$+ \frac{h_2}{k\sigma_0 \sqrt{Y_0^2 + Z_0^2}} \left[H_1 t_0 \cos(\omega_0 \theta - \alpha_0 - \Phi_0) + a I_0 \cos(\omega_0 \theta - \beta_0 - \Phi_0)\right] \quad \text{(I-27)}$$

or

$$q_2 = U t_m + V_0 t_0 \cos(\omega_0 \theta - \alpha_0 - \Phi_0)$$

$$+ \frac{U(a I_m - \xi' \Delta R_b)}{H_1} + \frac{V_0 a I_0}{H_1} \cos(\omega_0 \theta - \beta_0 - \Phi_0) \quad \ldots \quad \text{(I-28)}$$

where

$$U = \frac{1}{\frac{1}{H_1} + \frac{1}{h_2} + \frac{L}{k}} \quad \ldots \quad \text{(I-29)}$$

$$V_0 = \frac{H_1 h_2}{k\sigma_0 \sqrt{Y_0^2 + Z_0^2}} \quad \ldots \quad \text{(I-30)}$$

As stated earlier, the solution for the response to harmonics of the outdoor temperature and solar intensity wave will be of the same form as the second and fourth terms of (I-28) respectively, and may be superposed on the basic solution. The complete solution is then

$$q_2 = U t_m + [V_0 t_0 \cos(\omega_0 \theta - \alpha_0 - \Phi_0) + V_1 t_1 \cos(\omega_1 \theta - \alpha_1 - \Phi_1) + \ldots]$$

$$+ \frac{U(a I_m - \xi' \Delta R_b)}{H_1} + \frac{a}{H_1} [V_0 I_0 \cos(\omega_0 \theta - \beta_0 - \Phi_0)$$

$$+ V_1 I_1 \cos(\omega_1 \Phi - \beta_1 - \Phi_1) + \ldots] \quad \ldots \quad \text{(I-31)}$$

Finally, defining a decrement factor

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$$\lambda = \frac{V}{U} \quad . \quad . \quad .

In the present case, assuming a surface temperature of 70 F (at which temperature the saturation pressure of water vapor is 0.3628 lb per square inch, or 18.76 mm of mercury) the vapor pressure at 50 per cent relative humidity is:

$$P_v = (0.50)\left(\frac{18.8}{760}\right) = 0.0123 \text{ atmospheres}$$

Based on the statement of Hottel in an earlier paper[9] that $X = 1.8$ times the layer thickness, for layers bounded by parallel planes,

$$X = (1.8)\ (5000) = 9000 \text{ ft.}$$

Then

$$p_v X = (0.0123)\ (9000) = 112 \text{ atm.-ft.}$$

Referring to Fig. II-1 which has been mathematically extrapolated to large values of $p_v X$, it is found that R' for $p_v X = 112$ is equal to 105 Btu per hour-ft^2.

While the curve of Fig. II-1 continues to rise very slowly until $R' = 115$, it must be remembered that it is based on a 65 F average temperature and that extension to layer thicknesses including the colder upper atmosphere would require a downward revision of the average temperature with a consequent lowering of the curve. In any case, the value of $R' = 105$ is not likely to be exceeded by very much when the basic atmospheric temperature at the surface is 70 F.

The black body radiation from a square foot of surface at 70 F is, by the Stefan-Bolzman Law,

$$\begin{aligned} R_b &= 0.174 \times 10^{-8}\, T_1^4 \\ &= (0.174 \times 10^{-8})\ (530)^4 \\ &= 137 \text{ Btu per hour-ft}^2 \end{aligned} \qquad \text{(II-1)}$$

where T_1 is in degrees Fahrenheit absolute.

Hence the basic net outward radiation is

$$R_b - R_b' = \Delta R_b = 137 - 105 = 32 \text{ Btu per hour-ft}^2 \qquad \text{(II-2)}$$

If the average temperature of origin in the first 5000 ft were 70 F instead of 65 F, R' would be approximately 110 Btu per hour-ft^2 as determined by an extrapolation of Hottel's curves. Hence the layer's emissivity would be 110/137 or 0.8. Based on this,

$$F' = 0.8\ F \qquad \text{(II-3)}$$

APPENDIX III

HARMONIC ANALYSIS OF SOLAR INTENSITY AND OUTDOOR AIR TEMPERATURE CURVES

Any quantity that is a function of time and which varies in magnitude periodically in such a manner that each cycle of variation is identical with each other cycle can be expressed as a constant plus a series of sinusoidal variations of different frequencies. Such a series is called a Fourier series. Thus, if the function to be represented is

$$y = f(\theta) \qquad \text{(III-1)}$$

the Fourier series is

$$\begin{aligned} y = \bar{y} &+ M_0 \cos \omega_0\theta + N_0 \sin \omega_0\theta + M_1 \cos \omega_1\theta + N_1 \sin \omega_1\theta \\ &+ M_2 \cos \omega_2\theta + N_2 \sin \omega_2\theta + \ldots \end{aligned} \qquad \text{(III-2)}$$

In (III-2) ω_0, ω_1, ω_2, etc., are the angular velocities corresponding to frequencies of the fundamental and the various harmonics of the sinusoidal variations. Expressed in radians,

$$\omega_0 = \frac{2\pi}{\text{(time to complete one cycle of the fundamental)}}$$

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TABLE III-1—HARMONIC ANALYSIS OF INTENSITY OF SOLAR RADIATION ON HORIZONTAL SURFACE AT PITTSBURGH, PA., SEPTEMBER 10, 1931

1	2	3	4	5	6	7	8	9	10	11	12
TIME	y	$\omega_0 t$, DEG	$\cos \omega_0 t$	$y \cos \omega_0 t$	$\sin \omega_0 t$	$y \sin \omega_0 t$	$\omega_1 t$, DEG	$\cos \omega_1 t$	$y \cos \omega_1 t$	$\sin \omega_1 t$	$y \sin \omega_1 t$
12 M	0	0	1.000	0	0	0	0	1.000	0	0	0
1 A.M.	0	15	0.966	0	0.259	0	30	0.866	0	0.500	0
2 A.M.	0	30	0.866	0	0.500	0	60	0.500	0	0.866	0
3 A.M.	0	45	0.707	0	0.707	0	90	0	0	1.000	0
4 A.M.	0	60	0.500	0	0.866	0	120	−0.500	0	0.866	0
5 A.M.	0	75	0.259	0	0.966	0	150	−0.866	0	0.500	0
6 A.M.	10	90	0	0	1.000	10.	180	−1.000	− 10.	0	0
7 A.M.	20	105	−0.259	− 5.	0.966	19.	210	−0.866	− 17.	−0.500	− 10.
8 A.M.	50	120	−0.500	− 25.	0.866	43.	240	−0.500	− 25.	−0.866	− 43.
9 A.M.	105	135	−0.707	− 74.	0.707	74.	270	0	0	−1.000	−105.
10 A.M.	110	150	−0.866	− 95.	0.500	55	300	0.500	55.	−0.866	− 95.
11 A.M.	110	165	−0.966	−106.	0.259	28.	330	0.866	95.	−0.500	− 55.
12 N	110	180	−1.000	−110.	0	0	360	1.000	110.	0	0
1 P.M.	125	195	−0.966	−121.	−0.259	32.	30	0.866	108.	0.500	63.
2 P.M.	110	210	−0.866	− 95.	−0.500	− 55.	60	0.500	55.	0.866	95.
3 P.M.	75	225	−0.707	− 53.	−0.707	− 53.	90	0	0	1.000	75.
4 P.M.	50	240	−0.500	− 25.	−0.866	− 43.	120	0.500	− 25.	0.866	43.
5 P.M.	20	255	−0.259	− 5.	−0.966	− 19.	150	−0.866	− 17.	0.500	10.
6 P.M.	0	270	0	0	−1.000	0	180	−1.000	0	0	0
7 P.M.	0	285	0.259	0	−0.966	0	210	−0.866	0	−0.500	0
8 P.M.	0	300	0.500	0	−0.866	0	240	−0.500	0	−0.866	0
9 P.M.	0	315	0.707	0	−0.707	0	270	0	0	−1.000	0
10 P.M.	0	330	0.866	0	−0.500	0	300	0.500	0	−0.866	0
11 P.M.	0	345	0.966	0	−0.259	0	330	0.866	0	−0.500	0
	895.0			−714.0		27.0			329.0		− 22.0
$\bar{y} = 37.0$			$M_0 = -59.6$		$N_0 = 2.3$			$M_1 = 27.4$		$N_1 = -1.9$	

$\sqrt{M_0^2 + N_0^2} = 59.6$ | $\delta_0 = tan^{-1} \frac{N_0}{M_0} = 178$ | $\sqrt{M_1^2 + N_1^2} = 27.4$ | $\delta_1 = tan^{-1} \frac{N_1}{M_1} = 356$

Thus, $y = 37 - 59.6 \cos (15\theta - 178) + 27.4 \cos (30\theta - 356) + \ldots$ where y is the intensity of solar radiation (I). This expression is equation (20) in the main body of the paper.

and in degrees

$$\omega_0 = \frac{360}{\text{(time to complete one cycle of the fundamental)}}$$

The other angular velocities are related to the fundamental as follows:

$$\omega_1 = 2\omega_0$$
$$\omega_2 = 3\omega_0$$
$$\omega_4 = 4\omega_0$$
$$\text{etc.}$$

The harmonic series can also be written as follows:

$$y = \bar{y} + \sqrt{M_0^2 + N_0^2}\cos(\omega_0\theta - \delta_0) + \sqrt{M_1^2 + N_1^2}\cos(\omega_1\theta - \delta_1) + \quad . \quad . \quad \text{(III-3)}$$

where

$$\tan\delta = \frac{N}{M} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(III-4)}$$

and

$\sin\delta$ has the sign of N
$\cos\delta$ has the sign of M

Equation (III-3) is the form used in the main body of the paper to represent the variation in outdoor temperature and solar intensity.

In representing the outdoor air temperature and intensity of solar radiation on a surface it is assumed that these quantities have the same magnitudes at the same time of day on successive days. This is, of course, an approximation, but is sufficiently accurate for practical purposes. Furthermore, in any practical case it is rarely necessary to consider harmonics higher than the first, and often the fundamental plus the constant term gives a sufficient degree of accuracy, as pointed out in the main body of the paper.

The harmonic series representing a given variation in outdoor air temperature or solar intensity can be determined easily by carrying out the following steps. This procedure is illustrated by an example deriving the harmonic series for the solar intensity curve at Pittsburgh, Pa., on a horizontal surface, September 10, 1931,[2] as shown by Fig. 7a. Table III-1 gives the numerical example, which is referred to in the following step-by-step explanation.

Given a curve $y = f(\theta)$ over a period of one day:

1. Enter in the second column of the form shown in Table III-1 the hourly ordinates y of the curve to be analyzed.
2. In the third column, list the values of $\omega_0\theta$ for each hour, and in the fourth and sixth columns list the corresponding values of $\cos\omega_0\theta$ and $\sin\omega_0\theta$, respectively.
3. The fifth and seventh columns are filled in by multiplying each hourly ordinate y by the corresponding value of $\cos\omega_0\theta$ and $\sin\omega_0\theta$, respectively.
4. In the eighth column, list the values of $\omega_1\theta$ (which are twice as large as $\omega_0\theta$ at any given hour), and in the ninth and eleventh columns, respectively, list the corresponding values of $\cos\omega_1\theta$ and $\sin\omega_1\theta$.
5. The tenth and twelfth columns are filled in by multiplying each hourly ordinate y by the corresponding values of $\cos\omega_1\theta$ and $\sin\omega_1\theta$, respectively.
6. Add the ordinates in the second column and divide by 24; this gives the average ordinate of the curve, $\bar{y}$.
7. Add the values of $y\cos\omega_0\theta$ in the fifth column and divide the sum by 12; this yields the coefficient M_0 of the $\cos\omega_0\theta$ term in the final expression for the curve.
8. Add the values of $y\sin\omega_0\theta$ in the seventh column and divide the sum by 12; this yields the coefficient N_0 of the $\sin\omega_0\theta$ term in the final expression.
9. By repeating this procedure with the first harmonic tabulations in columns ten and twelve, coefficients M_1 and N_1 are obtained.
10. This method may be extended to any number of harmonics by setting up additional columns for $\omega\theta$, $\cos\omega\theta$, $\sin\omega\theta$, $y\cos\omega\theta$ and $y\sin\omega\theta$ in each harmonic desired.

The final expression for y is then given by (III-2). Whenever so desired, this can be reduced to the form of equation (III-3), determining the displacement angle δ from (III-4). This procedure is illustrated in Table III-1.

XUM

DISCUSSION

PROF. F. E. GIESECKE: This paper is a very valuable addition to the literature on air conditioning. The authors must have devoted much time and labor to its preparation, and I congratulate them. They state that the method is applicable only to homogeneous walls, but that they will expand it to apply also to non-homogeneous walls. I look forward with interest to their next report.

J. H. WALKER: This is a very excellent paper and the methods given should prove useful in calculating heat gain. The Research Committee has felt, however, that some actual tests should be made on a few types of walls, covering a wide range of capacities and conductivities. With those results as points of reference and with methods such as the paper describes of calculating the types of walls that lie, as to their thermal characteristics, between those tested, we can build up a complete set of tables or curves for THE GUIDE. It is high time that this disturbing lack of knowledge of heat gain through walls be remedied.

To this end we are planning to construct at our Pittsburgh Laboratory this spring two cubicles, each of which will have at least three different kinds of wall and roof construction, through which the heat gain will be actually measured.

No. 1124

ANALYSIS OF SUMMER WEATHER DATA IN THE UNITED STATES

By J. C. ALBRIGHT,* NEW YORK, N. Y.

IN modern necessities developed by the mechanical arts section of industry, there are many heat exchange and conversion problems in which atmospheric air is involved. The most common, although luckily not the most sensitive to the atmospheric conditions, is the automobile. The density of the air affects the proper mixture for combustion, the moisture content of the air varies the power obtainable from the individual fuel, while the temperature of the air affects the dissipation of heat from the radiator. Similarly, the operation of all other fuel burning apparatus for the production of mechanical or electrical energy is affected by the pressure, temperature and moisture condition of the atmosphere. Such fuel burning devices are not usually sufficiently sensitive to the atmospheric conditions to warrant readjustment hourly, or even daily, to meet changed conditions.

On the other hand, heating of buildings in the winter, and cooling of them in the summer, not only are affected by the changes in atmospheric conditions, but an immediate readjustment of the apparatus involved is required to compensate for them. The important variables for which correction must be effected are, in the order of their importance, wet-bulb temperature, dry-bulb temperature, dew-point temperature, wind velocity and sunshine. As the barometric pressure at any one location varies through only narrow limits, individual adjustment to compensate for changes in it is not required. Precipitation is of importance only in its resultant effect upon temperature and moisture content of the air. Specialized problems affected by the quality of the atmosphere include the production of textiles, printing, atmospheric water cooling, painting and food production, to name only a few.

In some of the cases mentioned, the correction of the apparatus to care for variable qualities in the atmosphere involves merely an adjustment of the equipment as originally furnished and the design would be the same regardless of the section of the country in which it was to be used. Automobiles and fuel burning equipment come under this classification. In other problems, it is necessary to design the equipment for the specific installation, adjusting the

* Vice-President, The Marley Company, Kansas City, Kan.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Pittsburgh, Pa., January, 1939.

XUM

Form No. 1030—Met'l.

U. S. DEPARTMENT OF AGRICULTURE, WEATHER BUREAU

MONTHLY METEOROLOGICAL SUMMARY

Station, PITTSBURGH, PA. *75th meridian time* *Month,* JULY, 1933

Date	Temperature (°F.) Thermometers over roof, 353 feet above ground							Moisture								Wind		Per cent of sunshine†	River stage, feet
	Extremes		Mean	Departure from normal	Dry bulb			Relative humidity, per cent			Vapor pressure, local noon	Precipitation, inches*				Velocity, miles per hour	Direction of max. velocity		
												Total		Snowfall, unmelted					
	Max.	Min.			8 a. m.	Local noon	8 p. m.	8 a. m.	Local noon	8 p. m.		8 a. m. to 8 p. m.	Mid. to mid.	8 p. m. to 8 p. m.	Depth on ground 8 p. m.	Max.			
1	89	73	81	+ 7	76	88	78	68	52	72	.684	.06	.06	0	0	32	n.	72	10.4
2	80	64	72	− 2	72	71	70	85	99	95	.732	1.23	1.23	0	0	18	ne.	9	10.4
3	74	59	66	− 8	62	66	71	74	70	48	.448	0	.01	0	0	32	e.	59	10.6
4	82	54	68	− 6	61	77	76	75	3[illegible]	37	.277	0	0	0	0	16	n.	96	11.5
5	81	58	70	− 4	64	77	76	67	33	35	.310	0	0	0	0	19	nw.	98	11.1
6	85	56	70	− 4	65	79	81	63	31	32	.298	0	0	0	0	15	w.	97	10.9
7	88	62	75	+ 1	71	85	83	54	31	31	.360	0	0	0	0	15	sw.	95	10.7
8	86	69	78	+ 4	74	86	75	52	34	66	.417	.03	.03	0	0	19	nw.	35	10.5
9	85	68	76	+ 2	72	80	80	79	63	87	.638	0	0	0	0	15	n.	71	10.3
10	84	61	72	− 2	66	80	80	77	37	29	.373	0	0	0	0	15	n.	82	10.3
11	85	62	74	− 1	69	80	80	51	31	23	.310	0	0	0	0	17	ne.	100	10.2
12	83	62	72	− 3	68	81	77	61	37	42	.387	0	0	0	0	15	e.	87	10.0
13	81	64	72	− 3	68	78	76	61	38	42	.373	0	0	0	0	20	se.	100	10.0
14	84	62	73	− 2	66	80	77	56	29	38	.298	0	0	0	0	17	se.	85	9.9
15	85	65	75	0	68	85	68	68	36	88	417	.34	.34	0	0	27	w.	27	10.0

16	80	62	71	− 4	65	78	70	90	59	76	.575	.41	.41	0	0	0	0	26	nw.	41	10.0
17	79	60	70	− 5	64	74	76	76	46	39	.402	0	T.	0	0	0	0	13	w.	99	10.0
18	83	60	72	− 3	67	79	75	62	42	44	.417	0	.02	0	0	0	0	15	sw.	68	10.0
19	89	63	76	+ 1	71	83	82	64	37	43	.442	0	0	0	0	0	0	15	sw.	96	10.2
20	91	65	78	+ 3	72	86	86	69	32	39	.402	0	0	0	0	0	0	14	sw.	100	10.2
21	94	71	82	+ 7	75	91	87	72	31	40	.446	0	0	0	0	0	0	14	sw.	81	10.1
22	96	66	81	+ 6	79	93	82	59	32	49	.490	.42	.42	0	0	0	0	41	nw.	61	10.1
23	98	71	84	+ 9	77	94	89	65	35	42	.575	0	0	0	0	0	0	18	s.	100	9.9
24	91	70	80	+ 5	77	88	73	66	48	88	.638	.30	.33	0	0	0	0	26	n.	55	9.8
25	79	68	74	− 1	70	75	73	90	80	78	.664	0	0	0	0	0	0	15	n.	1	10.2
26	70	67	68	− 7	67	70	68	71	74	84	.536	.04	.04	0	0	0	0	12	nw.	0	10.1
27	82	64	73	− 2	66	76	77	88	67	57	.595	0	0	0	0	0	0	10	sw.	36	10.2
28	87	63	75	0	68	81	82	84	52	43	.536	0	0	0	0	0	0	18	sw.	77	10.1
29	88	68	78	+ 3	74	78	85	82	73	60	.707	.35	.35	0	0	0	0	30	sw.	97	10.2
30	90	73	82	+ 7	76	86	85	76	59	55	.732	0	0	0	0	0	0	19	sw.	53	10.9
31	93	75	84	+ 9	77	90	86	83	49	57	.707	0	0	0	0	0	0	15	sw.	96	10.4
Mean		85.2	64.7	75.0	+0.4	69.9	81.1	78.2	70.6	43.4	51.9	.491	Total 3.18	Total 3.21	Total 0	Blank	Blk	Blk	Blk	Blk	Mean 70	Mean 10.3
Normal				74.6	Blank				75	54	60		Blank	4.05	0.0	Blank	Blk	Blk	Blk	Blk	64	

FIG. 1. MONTHLY WEATHER ANALYSIS AS PUBLISHED BY THE CLIMATE AND CROP WEATHER DIVISION OF THE U. S. WEATHER BUREAU

COMPARATIVE DATA AND EXTREMES

Month	Temperature							Precipitation						Relative humidity, per cent			Sunshine		Wind				Average number of days										Station Pressure + 2000 feet
	Average daily maximum	Average daily minimum	Average	Absolute highest	Year	Absolute lowest	Year	Average total	Greatest in 24 hours	Year	Average snowfall	Greatest snowfall in 24 hours	Year	8 a. m.	Noon—local time	8 p. m.	Total hours	Per cent	Average hourly velocity	Prevailing direction	Highest velocity	Direction highest velocity	Year	Clear	Partly cloudy	Cloudy	Rain, .01 or more	Snow (trace or more)	Thunderstorms	Dense fog	Max. temp. 90° or above	Minimum temp. 32° or below	Minimum temp. 0° or below	
Length of record, years	59	59	59	59	59	59	59	62	63	63	49	41	41	46	18	46	37	37	29	63	61	61	61	63	63	63	63	54	48	41	59	59	59	61
January	36.8	23.7	31.2	75	1906	−12	1875	2.97	2.34	1884	8.6	11.2	1914	79	68	72	95	33	11.9	w.	56	w.	1928	4	9	18	15	14	0	2	0	23	1	.129
February	39.9	23.8	31.8	77	1900	−20	1899	2.54	2.01	1887	7.5	11.0	1910	79	68	71	115	37	11.8	nw.	49	w.	1922†	4	9	15	14	13	0	1	0	21	0	.125
March	49.6	31.3	40.0	84	1929	1	1900	3.08	2.04	1908	5.7	15.0	1902	76	62	65	168	45	12.2	nw.	54	w.	1920	6	10	15	15	11	2	1	0	18	0	.119
April	60.6	41.3	51.0	90	1925	11	1923	2.96	2.60	1901	2.1	12.7	1901	72	56	60	198	50	11.6	nw.	53	w.	1909	7	10	13	13	4	3	1	0	5	0	.095
May	72.0	51.9	62.0	95	1887	27	1876	3.20	2.98	1889	T.	0.9	1923	72	55	59	257	57	10.2	nw.	53	nw.	1914	9	11	11	13	0	6	0	1	0	0	.083
June	80.1	60.5	70.3	98	1895	39	1879	3.67	3.19	1912	0	0	0	74	55	62	285	63	9.5	nw.	51	nw.	1917	7	14	9	12	0	8	1	3	0	0	.063
July	84.1	64.8	74.4	103	1881	49	1894	4.13	3.85	1887	0	0	0	75	54	60	297	65	8.9	nw.	46	nw.	1919	9	15	7	12	0	9	1	6	0	0	.109
August	82.0	62.9	72.4	103	1918	45	1890	3.19	3.65	1888	0	0	0	78	58	61	262	61	8.7	nw.	44	nw.	1925	10	13	8	10	0	7	2	4	0	0	.128
September	76.5	57.3	66.9	102	1881	35	1879*	2.55	4.06	1876	0	T.	1895	80	57	63	224	62	9.0	nw.	46	n.	1917	10	12	8	10	0	7	1	3	1	0	.152
October	64.4	45.8	55.1	91	1884	20	1887	2.50	2.22	1929	0.2	3.0	1917	80	59	64	187	54	9.8	nw.	47	nw.	1913	10	10	11	10	1	1	6	0	1	0	.121
November	50.7	35.8	43.2	79	1876	1	1929	2.33	2.13	1926	1.8	14.0	1913	76	66	68	118	40	11.3	w.	43	nw.	1927	6	9	15	12	7	0	2	0	10	0	.152
December	41.1	27.3	34.2	73	1885	−9	1880	2.75	2.44	1901	6.4	10.7	1894	78	71	72	84	29	11.5	w.	54	w.	1909	4	9	18	14	12	0	2	0	21	0	.173
Year	61.6	43.9	52.8	103	Aug. 6 1918	−20	Feb. 10 1899	35.92	4.06	Sept. 17 1876	32.3	15.0	Mar. 5 1902	77	60	65	2294	50	10.6	nw.	56	w.	Jan. 19 1928	86	131	148	150	68	40	22	15	99	2	.140

*Also in 1895 and 1928 †Also Feb. 14, 1923. Wind is in True Velocity. Elevation of Barometer is 842 feet.

FIG. 2. COMPARATIVE WEATHER DATA FOR PITTSBURGH, PA., AS PUBLISHED AT THE END OF 1933

initial size or character of the apparatus so that it will perform acceptably under the extremes of conditions to be expected, while keeping the investment to the minimum possible figure. A simple example is that one would not buy a fur-lined overcoat in preparation for a winter in Florida.

Since the variations in the different weather factors affecting such designs are great, universal rules for design applicable over the entire country are impossible. Furthermore, since a variation in one criterion is not accompanied by a proportional change in others for the different sections of the country, knowledge of the expectancy in one factor does not permit fixing the values of the other variables to be used in a design. As an instance of the first statement, wind velocities of 120 mph have been experienced in certain sections of the country, yet it would be impractical to build all structures capable of withstanding such a velocity. On the second premise, air conditioning equipment designed for a theater in Cleveland need not be of as large capacity as for the same size theater in New Orleans, yet, if the designs were based only on the dry-bulb temperatures, the same installation would be made in both.

Values of each of the factors characteristic of that location must be used in determining a design, these values to be based upon a forecast of their probable recurrence during the life of the equipment. It is the purpose of this paper to analyze the expectancy of occurrence and characteristics of those variables in weather for the summer season that are of particular importance in design, with a view to furnishing the designer with a more satisfactory understanding and measure upon which to base the design, thus permitting the optimum result to be obtained. The bulk of the analysis is devoted to temperatures as being most important in present design practice, and the dry-bulb, wet-bulb and dew-point temperatures are considered simultaneously because of their interrelation. Wind analysis and solar radiation are treated separately as fully as their present usage in design warrants.

TEMPERATURE ANALYSIS

Sources of Data

Available for analysis of the expectancy in weather data are records obtained by both the Climate and Crop Weather, as well as the Aerological Divisions of the United States Weather Bureau. The Climatological records are published principally for the benefit of the public. The Aerological records are utilized for immediate weather analysis, to permit bulletins to be furnished to the aviation industry, but no accumulations of records are published by this division.

The best known of all the publications of the Climate and Crop Weather Division is the monthly summation of weather data, Meteorological Form No. 1030, such as is shown in Fig. 1 for July, 1933, in Pittsburgh. This information is assembled and published locally at approximately 200 stations located in cities throughout the United States. From it can be obtained daily data on the temperatures at noon local time and at 7:30 a.m. and p.m., 75th Meridian time. Note that since 75th Meridian time is Eastern Standard Time, this means that the morning and evening readings are at 4:30 o'clock in San Francisco, 6:30 in Chicago, and 5:30 in Denver. Thus only the noon readings

XUM

Form No. 1145—Aer.

U. S. DEPARTMENT OF AGRICULTURE, WEATHER BUREAU

MONTHLY SUMMARY OF TEMPERATURE AND DEW POINT

Station Pittsburgh Altitude 1250 Ft. Month July 193 3 Time 32 min. after hour 75th meridian

DAY	1	2	3	4	5	6	7	8	9	10	11	NOON	1	2	3	4	5	6	7	8	9	10	11	MID'T	SUM OF DIF.	MEAN DIF.
1	76	74	73	74 68	72	73	74	76 46	79	82	84	86 71	90	89	78	73 67	72	72	72	74 70	75	74	73	71 68	44 ✓	7.3
2	70	70	69	69 68	68	68	69	71 63	75	78	79	76 71	71	72	74	73 70	73	73	68	68 67	68	68	67	68 62	15 ✓	2.5
3	63	60	59	58 54	57	57	57	58 54	59	60	62	63 53	65	68	69	69 49	71	71	69	67 50	65	64	62	61 60	62 ✓	10.3
4	59	58	58	56 53	55	55	58	63 50	66	69	71	71 52	75	75	76	77 50	77	76	76	78 54	72	69	66	65 56	92 ✓	15.3
5	63	61	60	57 53	57	58	61	66 52	69	72	74	76 55	77	77	80	80 52	79	79	77	73 53	71	69	68	67 53	103 ✓	17.2
6	65	66	66	61 51	60	58	63	68 53	71	74	76	78 53	80	82	83	83 56	83	82	82	77 58	74	73	72	71 55	114	19.0
7	67	66	65	63 55	62	62	64	70 60	75	79	81	84 63	85	86	87	87 60	86	86	83	79 58	76	75	74	73 58	102	17.0
8	72	69	67	66 58	65	65	65	70 61	75	79	83	86 62	88	84	83	77 59	74	70	72	72 66	71	71	71	71 67	69 ✓	11.5
9	71	71	71	70 65	70	70	71	73 66	75	78	80	78 66	78	80	83	85 55	84	82	81	77 52	74	72	70	68 55	92	15.3
10	67	66	65	64 55	64	63	63	67 57	72	77	79	80 50	78	81	82	80 48	81	81	80	76 48	72	71	70	70 51	128 ✓	20.8
11	66	65	63	62 58	61	60	63	68 52	72	75	76	78 50	79	80	83	82 51	83	82	81	76 49	72	72	69	69 50	130	24.7
12	68	64	62	62 53	62	61	63	68 56	71	74	76	78 53	79	81	80	81 52	80	79	77	74 50	71	70	68	67 52	112 ✓	18.7
13	66	64	63	62 55	62	61	63	67 55	68	72	75	78 54	76	78	77	81 54	79	76	75	73 50	69	67	65	64 54	100 ✓	16.7
14	64	62	62	59 59	58	59	61	65 57	69	73	75	79 50	79	80	82	80 54	81	78	76	73 51	70	69	68	66 54	108 ✓	18.0
15	65	63	62	62 58	63	62	64	68 60	70	74	76	80 62	83	83	68	78 60	68	66	67	66 68	66	66	67	68 46	47 ✓	7.8
16	63	63	63	62 59	62	62	62	64 62	67	71	77	77 64	76	63	67	69 63	68	69	69	66 62	65	64	63	62 44	30 ✓	5.0
17	62	62	60	59 58	59	60	62	64 56	68	70	71	73 59	74	76	76	78 44	78	77	76	73 54	70	69	66	67 53	94 ✓	15.7
18	65	62	62	60 49	59	58	62	65 58	70	74	77	78 53	78	78	79	80 58	80	81	78	74 61	70	69	69	68 56	93	15.5
19	67	66	65	64 57	64	63	66	70 60	76	79	81	82 55	84	85	84	85 55	87	84	82	78 56	76	75	74	73 57	112 ✓	18.7
20	70	69	67	67 61	67	67	67	72 58	78	83	84	86 62	87	89	89	91 55	87	87	85	83 60	81	78	76	74 46	126 ✓	21.0
21	72	76	70	68 63	68	67	69	74 65	78	81	85	89 60	91	92	92	93 57	92	91	89	86 60	85	82	80	80 60	125 ✓	20.8
22	77	77	75	75 61	73	72	73	78 66	82	86	88	93 61	89	94	94	72 62	75	79	78	76 65	76	75	75	75 63	93	15.5
23	74	72	72	71 63	76	69	71	76 67	81	84	88	92 30	94	95	96	96 68	96	95	89	85 46	83	81	79	78 65	99	16.5
24	77	76	77	78 65	79	72	73	76 67	78	80	83	86 69	88	88	88	82 70	70	71	72	71 70	71	70	70	70 68	54	9.0
25	69	68	68	67 66	67	67	67	68 67	69	72	72	73 68	74	76	76	76 67	76	75	72	71 67	70	69	68	68 65	23	3.8
26	67	67	66	65 62	65	64	65	65 59	65	66	66	66 60	67	65	63	63 63	63	65	65	65 63	65	64	64	64 62	19	3.7
27	64	63	63	63 62	63	64	64	66 63	70	74	78	78 63	78	79	79	79 64	79	77	76	74 66	73	72	70	68 63	48	8.0
28	67	66	66	66 62	64	64	66	72 65	75	77	78	79 65	83	85	86	86 64	86	85	82	78 66	77	75	73	72 64	65	10.8
29	71	69	68	67 63	67	68	71	74 67	76	80	82	82 74	81	85	85	87 72	87	87	84	81 68	79	78	76	75 70	52	8.7
30	75	75	73	73 68	73	73	79	75 69	74	77	82	84 79	86	88	89	87 67	85	88	85	82 70	81	80	80	77 70	66	11.0
31	79	77	76	75 68	74	74	75	78 73	81	84	85	87 69	90	90	92	92 70	91	89	86	84 68	82	80	79	78 70	78	12.6
Sums	2131	2081	2056	2030 1819 211	2007	1996	2046	2125 2005 120	2354	2353	2410	2472 2302 170	2503	2503	2514	2498 2328 170	2471	2453	2409	2324 2174 150	2270	2231	2172	2126 1966 160	2497	415.3
MEANS	68.4	67.1	66.3	65.5 64.8 67	64.7	64.4	66.0	68.5 66.7 1.8	75.9	78.1	77.7	79.7 79.1 19.5	80.7	80.7	81.3	80.5 79.5 21.1	79.7	79.1	77.5	75.0 72.8 15.0	73.3	72.0	70.1	68.0 66.0 17	80.4	13.4
Mean temperature				67				68																	80.3	13.4

FIG. 3. MONTHLY DRY-BULB AND DEW-POINT TEMPERATURES AS ASSEMBLED BY THE AEROLOGICAL DIVISION OF THE U. S. WEATHER BUREAU

of the Climate and Crop Weather Division are at uniform local time and of value in comparing cities in different parts of the country.

In addition to the monthly summary, each of the Climatological Stations issues an annual summary containing the monthly averages of the various factors for the year; this report also contains a summation of the monthly averages on each factor for all the years of record as shown in Fig. 2. Most of the stations have been taking records for a period of over 20 years, many of them for 50 years, hence these data are very valuable for checking to see whether a particular year has been an average year, or has gone towards either extreme on the various factors.

Since average values of one or two daily readings only are not satisfactory for the determination of proper design values in the heat exchange industry, very little basic data in readily usable form can be gleaned from the records available in the Climate and Crop Weather Division. The Aerological Division, while it does not publish the data collected, does take temperature readings at hourly intervals at its stations in the various airports throughout the country. The dry-bulb temperature and the wet-bulb temperature are read each hour, the dew-point is calculated from these readings, then the dry-bulb and the dew-point are recorded on a daily sheet, which data remain on file at the station. No monthly or other periodic summation is made of these readings, either for publication or department use. Luckily, however, there was a period of 5 years, from 1930 to 1934, inclusive, when a monthly summary sheet was made and a pencil copy forwarded to Washington. A section of such a form, with the readings inserted for Pittsburgh, is shown in Fig. 3, and cover the same month as in Fig. 1. From these data, the dry-bulb and dew-point temperatures may be analyzed hour by hour, also the wet-bulb temperatures can be calculated from these readings, thus enabling a count to be made of the number of times each temperature has occurred in a period at several individual hours of the day.

By arrangement with D. M. Little, in charge of the Aerological Division, photostatic copies were obtained of these monthly summaries from practically all of the stations for which they were on file, and for as many years as they were available. The year of 1932 was not available from any of the stations, as the records for that year had been found in volumes from which it was impractical to attempt to obtain copies. The resulting mass of data for this study, therefore, included three or four years of records from 67 cities scattered as uniformly as possible over the United States. On all of the records, the readings of dew-point in the early years had been taken at four-hour intervals only, and in some cases six hours, but in later years all records contained 24 readings per day.

FIG. 3. MONTHLY DRY-BULB AND DEW-POINT TEMPERATURES AS ASSEMBLED BY THE AEROLOGICAL DIVISION OF THE U. S. WEATHER BUREAU

Summation of Summer Temperature Data

The records containing six readings per day, or more, fitted nicely into the plans for developing weather data which were to be based on an analysis of conditions at four-hour intervals, each reading to be considered as representative of the two hours preceding and the two hours following. When the fact is considered that the Weather Bureau has found that the average of the daily maximum and minimum dry-bulb temperatures at any one station comes within less than a degree of the 24-hourly average, readings at four-hour

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TABLE 1—ANALYSIS OF WET-BULB
Frequency of Occurrence During Season of June to

WET-BULB TEMP. F	AT TEMPERATURES SHOWN NUMBER OF READINGS AT TIME INDICATED						Total Hours per Season
	A.M.			P.M.			
	3	7	11	3	7	11	
79	...	...	...	6	6	...	12
78	...	...	4	5	3	2	14
77	...	...	5	12	7	2	26
76	1	1	8	19	11	9	49
75	2	5	30	35	29	15	116
74	8	6	23	32	29	21	119
73	20	16	49	43	46	29	203
72	19	24	41	45	43	34	206
71	37	29	41	46	51	36	240
70	38	28	42	39	37	58	242
69	45	34	24	21	26	43	193
68	41	46	33	25	29	26	200
67	32	41	21	18	20	26	158
66	22	28	12	18	22	30	132
65	25	23	23	21	21	20	133
64	23	20	26	19	19	16	123
63	24	31	15	17	17	16	120
62	19	27	13	15	10	14	98
61	14	21	12	6	13	19	85
60	12	7	15	10	7	9	60
59	28	14	9	5	6	15	77
58	10	15	9	8	8	6	56
57	9	11	7	5	5	5	42
56	9	5	4	2	6	5	31
55	7	10	4	5	4	5	35
54	7	9	2	3	6	6	33
53	7	8	1	3	1	6	26
52	3	6	4	2	2	7	24
51	10	5	5	1	1	1	23
50	...	1	4	...	...	2	7
49	2	5	...	...	...	1	8
48	8	1	...	1	1	3	14
47	2	1	...	...	...	...	3
46	2	2	1	1	...	...	6
45	1	3	...	...	2	...	6
44	...	2	...	...	...	...	2
43	...	1	1	...	...	...	2
42	...	...	...	...	...	...	...
41	...	1	...	...	...	1	2
40	1	...	...	...	...	...	1
39	...	...	...	...	...	...	...
38	...	1	...	...	...	...	1
Avg Temp F	64.7	64.4	67.8	69.0	68.5	67.0	66.9
Col. No. 1..	2A	2B	2C	2D	2E	2F	3

NOTE: These Data taken from Form No. 1140 of the

TEMPERATURES, KANSAS CITY, MO.
September, Inclusive, in Years 1930–1931–1933–1934

AT OR ABOVE TEMPERATURES SHOWN NUMBER OF READINGS AT TIME INDICATED							
A.M.			P.M.			Total Hours per Season	Per Cent At or Above
3	7	11	3	7	11		
...	...	...	6	6	...	12	0.4
...	...	4	11	9	2	26	0.9
...	...	9	23	16	4	52	1.8
1	1	17	42	27	13	101	3.4
3	6	47	77	56	28	217	7.4
11	12	70	109	85	49	336	11.5
31	28	119	152	131	78	539	18.4
50	52	160	197	174	112	745	25.4
87	81	201	243	225	148	985	33.6
125	109	243	282	262	206	1227	41.8
170	143	267	303	288	249	1420	48.5
211	189	300	328	317	275	1620	55.3
243	230	321	346	337	301	1778	60.7
265	258	333	364	359	331	1910	65.3
290	281	356	385	380	351	2043	69.7
313	301	382	404	399	367	2166	74.0
337	332	397	421	416	383	2286	78.1
356	359	410	436	426	397	2384	81.4
370	380	422	442	439	416	2469	84.3
382	387	437	452	446	425	2529	86.4
410	401	446	457	452	440	2606	89.1
420	416	455	465	460	446	2662	91.0
429	427	462	470	465	451	2704	92.4
438	432	466	472	471	456	2735	93.4
445	442	470	477	475	461	2770	94.6
452	451	472	480	481	467	2803	95.8
459	459	473	483	482	473	2829	96.7
462	465	477	485	484	480	2853	97.3
472	470	482	486	485	481	2876	98.3
472	471	486	486	485	483	2883	98.6
474	476	486	486	485	484	2891	98.8
482	477	486	487	486	487	2905	99.2
484	478	486	487	486	487	2908	99.4
486	480	487	488	486	487	2914	99.5
487	483	487	...	488	487	2920	99.7
487	485	487	...	...	487	2922	99.8
487	486	488	...	...	487	2924	99.8
487	486	...	...	...	487	2924	99.8
487	487	...	...	...	488	2926	99.9
488	487	...	...	...	...	2927	99.9
...	487	...	...	...	...	2927	99.9
...	488	...	...	...	...	2928	100.0
...	...	...	...	...	...	....	....
4A	4B	4C	4D	4E	4F	5	6

Aerological Weather Station at Kansas City, Mo. Airport.

intervals would permit summations as accurate for the purposes of this study as hourly readings.

The first step in the analysis consisted of counting the number of times each dry-bulb, wet-bulb and dew-point temperature occurred at each of the six daily readings each year at each station during the season of June, July, August and September. The dry-bulb and dew-point values could be taken directly from the forms, but to obtain each wet-bulb temperature it was necessary to convert back from the dry-bulb and dew-point temperatures to obtain each individual reading. The conversion of over 150,000 such pairs of readings was the largest single task of the entire project.

The data obtained were summed up for the entire period on each city, and accumulated in a uniform condensed form similar to that shown for the wet-bulb temperature of Kansas City in Table 1. Columns 2A, 2B, 2C, 2D, 2E and 2F show the actual number of times that the various temperatures occurred at the four-hour intervals. The total number of readings at each temperature in the four years is given in Column 3. It is obvious that the total number of readings on four-hour intervals in a four-year period is, when each reading is considered representative of a four-hour period, also the average number of hours per season. Thus Column 3 is both total readings and average number of hours per season. Columns 4A to 4F, inclusive, are the accumulative totals of the readings in Columns 2A to 2F, respectively, or are the number of readings in which the temperature shown was equalled or exceeded. The total Column 5 is the number of readings in the four seasons, or hours per normal season, and the final column the percentage of time that each temperature was equalled or exceeded.

Summations similar to this one were made of the dry-bulb, wet-bulb and dew-point temperatures for the entire group of 67 cities. To amplify further the knowledge of the variations of these factors throughout the United States, and to check with the data from the Aerological Section, the average monthly noon dry-bulb, wet-bulb and dew-point temperatures for 1930 to 1934, inclusive, as taken at the Climatological Stations, were obtained for the 200 available locations, along with the averages of the morning and evening observations from the Climatological Stations for the 67 cities from which the Aerological data were taken. Comparison between the averages of readings taken simultaneously at Climatological and Aerological Stations in the same cities showed normal agreements within tenths of degrees.

STUDY OF COMPARATIVE CHARACTERISTICS

Temperatures do not remain fixed for even a short period of time, but are continually changing, both daily and seasonally, these variations having entirely different values and rates of change in different parts of the country. Since the equipment designed upon the basis of these accumulated data must have sufficient capacity to meet, with reasonable results, the normal maximum temperatures at the location, and flexibility enough to meet both the daily variation and the total range of the season, these characteristics deserve close study.

The total amount of the seasonal variation is obvious from an inspection of each summary sheet. The character of the distribution of the readings over

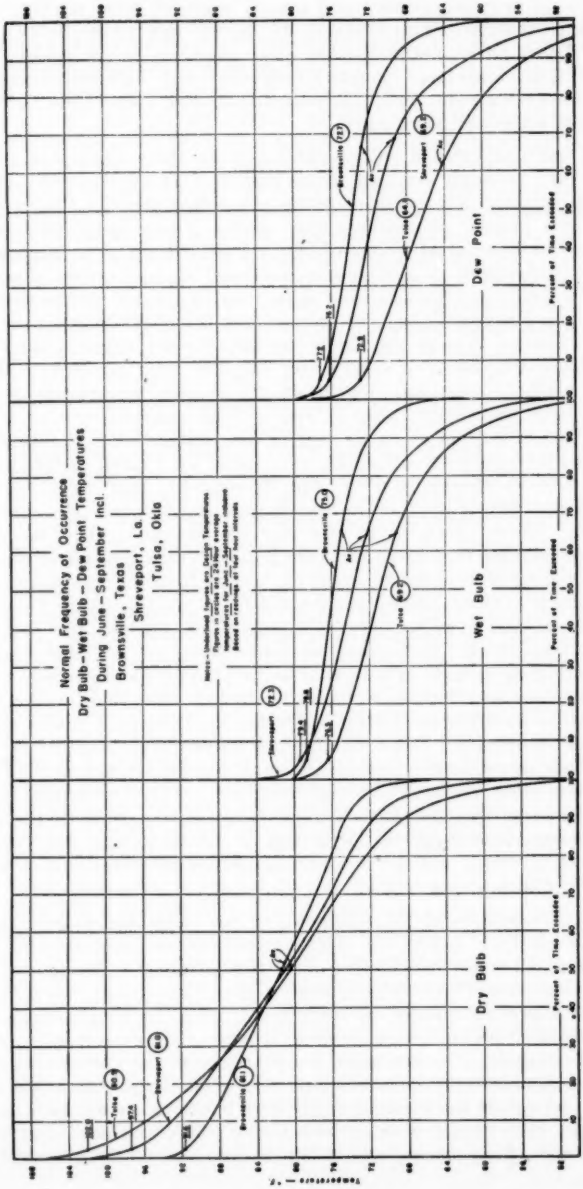

FIG. 4. COMPARATIVE SEASONAL TEMPERATURES AT REPRESENTATIVE LOCATIONS

this range is of more importance and can be observed most readily from a graph of the percentage of time that each of the readings is exceeded, as shown in Fig. 4. Lying on approximately the same meridian, in the middle of the western plains, far from large mountain ranges or other disturbing topography, the data from Brownsville, Tex., Shreveport, La., and Tulsa, Okla., afford an excellent basis for comparative study.

From the left-hand chart it will be noted that the three cities have approximately the same average dry-bulb temperature. This average was obtained by summing up all readings taken during the entire period of analysis and dividing by the number of readings, thus giving a true weighted average—(see averages at bottoms of Columns 2A to 3 in Table 1). It happens that the average temperature in each case is exceeded approximately 50 per cent of the time, a condition which does not always obtain. Yet it will be noted that the slope of the Tulsa curve is much steeper than that of Shreveport, which in turn exceeds that of Brownsville, the flatter slope of the southern cities being due to the blanketing effect of more water vapor in their atmosphere than in that of the northern locations. Obviously, the average temperature is not a satisfactory basis for design, and, in view of the readings, neither could a common number of degrees above the average temperature added to that average give a reasonable result. For instance, if a value were taken of 94 F, which is the maximum temperature noted for Brownsville, that value would be exceeded 7.5 per cent of the time in Shreveport, or 220 hours per season, and 381 hours in Tulsa, which is 13 per cent of the time. The use of the maximum temperature read in Brownsville in design would result in the specifying of equipment that would be operating at less than its rated capacity for the entire season in Brownsville; would possibly be satisfactory in Shreveport, where fly wheel action might carry the installation through the peaks that exceed that temperature; and would be found short of its requirements in Tulsa for such long periods at a time as to be entirely unsatisfactory. Obviously, a higher dry-bulb design basis must be used in Tulsa than in Shreveport, and Brownsville should be the lowest of the three. Perhaps it will be surprising to find that the Gulf Coast cities, even though they require the lower design dry-bulb temperatures, have, in general, higher average dry-bulb temperatures than more northern midwest locations. This apparently inverted condition, caused by the continuous decline in moisture content of the atmosphere from south to north, persists to a peak in central Kansas, and from that point north is reversed, and the dry-bulb design temperatures fall as normal judgment would dictate they should.

A similar investigation of the wet-bulb and dew-point temperatures reveals that on both of these factors the Tulsa record shows expectancy of lower temperatures at all times than at the other two locations. Shreveport, while normally having a lower wet-bulb temperature than Brownsville, has higher top readings and a greater number of readings at these top temperatures, hence Shreveport should have equipment designed to meet a higher wet-bulb temperature than Brownsville. On dew-point, the character of the curves is such that Brownsville must obviously have a higher dew-point design temperature than Shreveport. Note that the averages of all three locations are exceeded between 60 and 70 per cent of the time on both the wet-bulb and dew-point curves. Reasons for the average being not at the 50 per cent point are readily dis-

cernible from the shape of the curves, which are much flatter above the average point than below, thus showing that the maximum is fewer degrees above the average than the minimum is below it.

These seasonal time-temperature graphs, while similar in general contour throughout the country, vary greatly in the slope of the center portion of the curve, and in the percentage at which the ends begin to curve rapidly. The total range and slope of the graph varies from a minimum in Florida and Southern California, to a maximum in the central Rocky Mountain District, the values increasing along each coast from south to north, with those on the East Coast increasing more swiftly than on the West. Mid-continent curves are steeper than those on either coast at the same latitude. The widest dry-bulb and wet-bulb ranges are in Nevada. Arizona, notoriously arid, shows the greatest seasonal dew-point differential. These rather orderly variations in seasonal variations are an inverse function of the dew-point temperature, which fact *is of major importance in the determination of proper design temperature values.*

The daily variation to be expected is of perhaps even more importance in the design of heat exchange equipment than the anticipated seasonal range. Characteristic daily variations for a number of different cities on the dry-bulb, wet-bulb and dew-point temperatures are shown in Fig. 5. Since, from the previous discussion of Fig. 4, it is obvious that designers can make little use of average temperatures, but must base their work on some temperature nearer the maximum, this series of graphs show the temperatures which will be exceeded not more than six times per year. These six occasions did not, of course, all occur on the same six days, but the curves simply mean that the temperatures shown at 6 a.m. and 6 p.m. each would be exceeded six times in a normal year, although none of the high readings at 6 p.m. necessarily took place on the same day that a high reading was experienced at 6 a.m.

From repeated comparisons it was found that the shape of these curves checks directly with the shape of the weighted average temperature curves plotted for the various hours of the entire day; also that the seasonal range at any one hour is approximately equal to that of any other hour of the day. Such difference in range as exists at different hours tends to show a slightly greater seasonal variation in the afternoon dry-bulb than in the period just before dawn, but the exact reverse is true on the wet-bulb temperature, while the dew-point has no uniform trend.

The daily march of the dry-bulb follows the expected path, in that the temperature slides downward in a smooth curve throughout the night, reaching a minimum just about sunrise, turns rather abruptly upward and rises at a rapid rate that slows down as the day progresses, coming to a peak in the afternoon, and reversing the morning curve as it drops off. The evening cooling is not as rapid as the morning heating, and the curve slows down in its rate of drop from sundown until the minimum is reached the next morning.

Note that Seattle, Wash., despite a lower average dry-bulb temperature, should have a design dry-bulb value well above that for Laguna Beach, Calif., also that Wichita, Kan., with approximately the same average dry-bulb temperature as Miami, Fla., justifies on this graph a design value 13.5 F higher than Miami, whereas New York should have the same design basis as Miami, although its average dry-bulb temperature is 7.3 F lower. Similar inverted

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evidence is the rule rather than the exception throughout the country, again the direct result of relative moisture content in the atmosphere.

Other things being equal, the greater the amount of water vapor in the

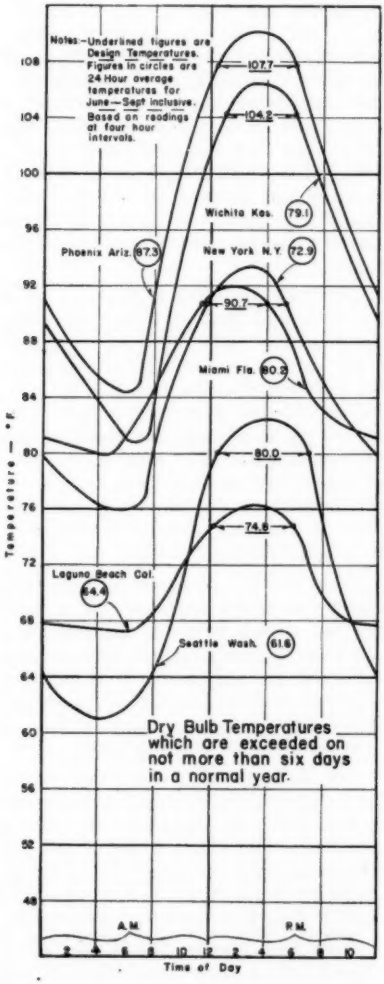
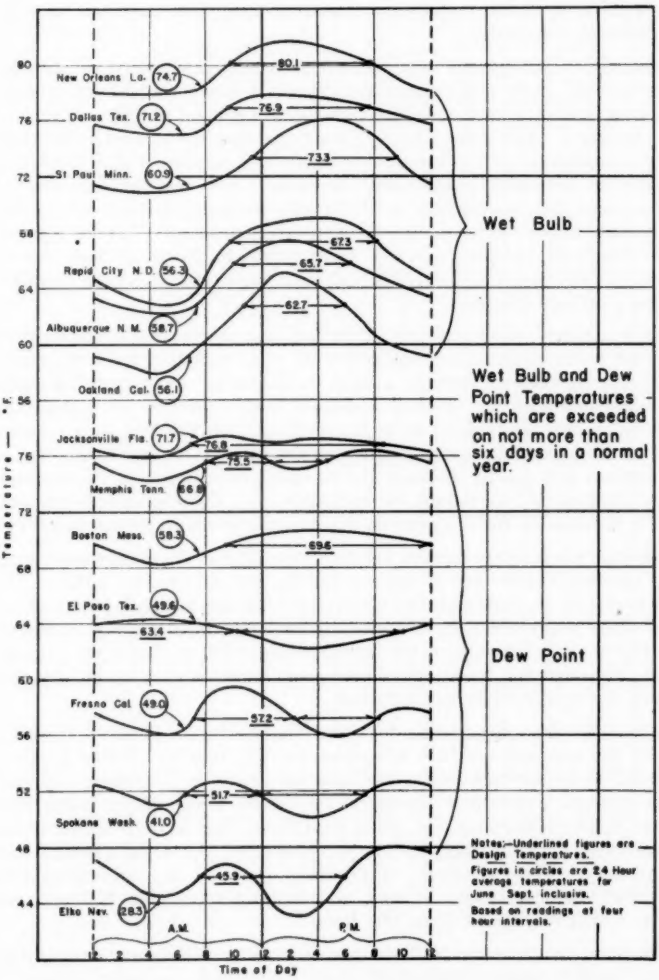

Notes.—Underlined figures are Design Temperatures. Figures in circles are 24 Hour average temperatures for June—Sept inclusive. Based on readings at four hour intervals.
107.7
104.2
Wichita Kas. 79.1
Phoenix Ariz. 87.3
New York N.Y. 72.9
90.7
Miami Fla. 80.2
80.0
74.8
Laguna Beach Cal. 64.4
Seattle Wash. 61.6
Dry Bulb Temperatures which are exceeded on not more than six days in a normal year.
Temperature — °F
Time of Day
A.M.
P.M.

FIG. 5. COMPARATIVE DAILY TEMPERATURE CHARACTERISTICS

(*Continued on opposite page*)

air, the smaller will be the daily and seasonal range of dry-bulb temperatures. One reason for this is that the water vapor in the air reduces both the radiant energy of the sun reaching the earth's surface during the day and also the

radiation to outer space at night. An important factor in controlling dry-bulb temperature at the surface is evaporation which process absorbs sensible heat, and thus tends to hold the maximum daily surface temperature down. At the



minimum temperatures, more dew is deposited in the areas having the higher water content in the atmosphere, and the condensing of this water vapor releases latent heat, thus tending to keep the minimum temperatures higher.

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These facts combine to account for the relatively small daily temperature ranges at locations having a normally high dew-point temperature. Most important, however, the reduction in dry-bulb temperature which normally accompanies an increase in dew-point temperature *precludes the probability of extreme dry-bulb temperatures occurring simultaneously with extremes in dew-point or wet-bulb temperature.*

The conductivity and specific heat, or heat absorbing capacity of the terrain, likewise affects the daily temperature variation. A damp surface has greater heat absorbing power than a dry one. Cultivated ground absorbs more heat than a sandy desert. The higher the absorbing qualities, the smaller is the daily swing in temperatures. The direction, velocity and source of the wind is a major factor in the daily temperature variation. For instance, air coming from warm areas and passing over heated plains increases in temperature and adds to the natural sun effect on the ultimate areas reached. The increase in number of hours of sunshine per day from South to North in the summer season tends to increase the differential between night and day time temperatures in the Northern locations.

There is less water vapor present on the average at elevated locations than near sea level hence there is less blanketing of solar radiation by water vapor in the air, and conversely, there is a large radiation to outer space at night, with the result that both the seasonal and daily ranges increase with elevation. Mountainous regions are subject to great mechanical turbulence in air movement, and as high wind velocities accelerate the rate of moisture evaporation from the surface and rapidly dissipate the moisture so evaporated, the tendency is to large temperature variations in such districts. Convection air currents are partially responsible for the extreme daily ranges noted in mountain valleys.

The wet-bulb temperature curves follow those of the dry-bulb temperature in general contour, but are not as smooth and the range is much smaller. The afternoon peak may be reached either before or after the dry-bulb peak and the crowns of these peaks are flatter. On individual days, the wet-bulb temperature may rise steadily throughout the day into the night, so that high wet-bulb temperature readings are frequently observed at night, whereas high dry-bulb temperature readings at night are very rare.

Logical design values for various locations frequently are inverted from the evidence of the average wet-bulb temperatures, for reasons similar to those already found to be the case in the dry-bulb temperatures. Albuquerque, with the higher average wet-bulb temperature, requires a lower design value than Rapid City, N. D. Oakland, Calif. (San Francisco), has a 5.1 F lower design value than Rapid City, although it has approximately the same average wet-bulb temperature. The magnitude of the daily range is not accompanied by a proportional seasonal range, as may be noted by comparing Oakland and Rapid City, which reverse on these two items.

The dew-point temperature is the surprising factor, following daily cycles that at first glance appear illogical. In most inland cities, a valley occurs in the dew-point graph during the afternoon, when the dry-bulb and wet-bulb temperatures normally reach their peaks. Climbing back to a peak in the evening, the dew-point begins to drop in the late evening, reaching a second valley coincidentally with the valleys of the dry-bulb and wet-bulb in the early

morning, rising to a peak before noon, then sliding rapidly down to the afternoon valley. Thus the characteristic dew-point graph shows a daily double cycle for most of the locations in the country. In seaboard cities the afternoon valley may disappear or be present to a minor degree. In the Western plains

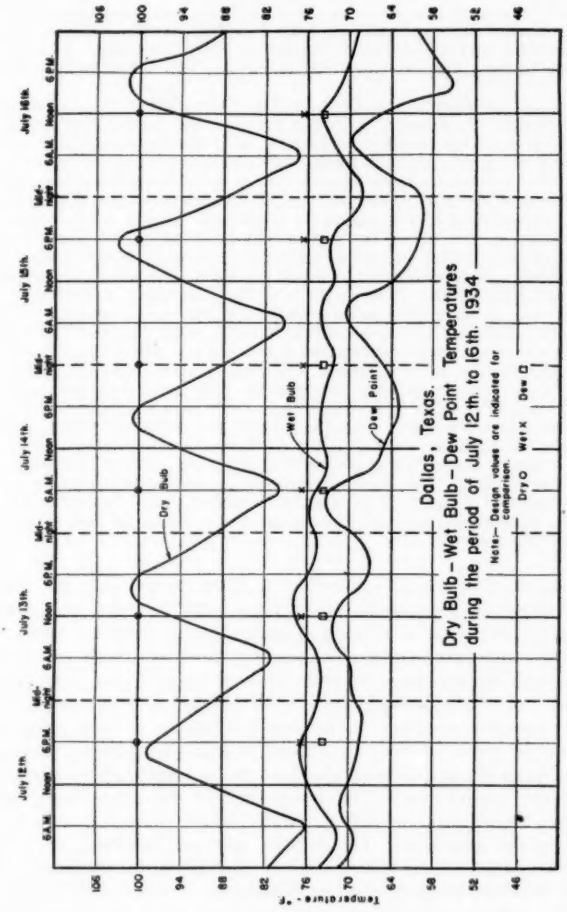

FIG. 6. TYPICAL TEMPERATURES AT DALLAS, TEXAS

and mountains, the morning valley is frequently washed out, in some cases the peak of the day occurring simultaneously with the minimum dry-bulb. Combinations of topographical features can give almost any single or double cycle characteristic daily curve for the dew-point temperature. L. P. Harrison,

Meteorologist of the U. S. Weather Bureau, comments on the behavior of the dew-point as follows:

"As is well known, the diurnal march of vapor pressure at the surface generally shows a regular periodic variation. Inland regions in summer show two maxima and two minima, occurring at about 6 to 9 a.m. and 8 to 9 p.m., for the former, and 3 to 4 p.m. and 3 to 4 a.m., for the latter. In general, the oceans in summer and winter and most inland regions in winter show but one maximum and one minimum, similar to the diurnal march of the dry-bulb temperature, the maximum occurring during the afternoon and the minimum during the early morning. Coastal stations show variations between the extremes outlined above, but resemble the oceans most closely.

"The causes of this diurnal march of absolute humidity at the surface are substantially as follows: In *summer*, at *inland* stations, the ground at dawn is greatly cooled due to the nocturnal radiation, especially so if the night has been celar. The subsidence of the air during the night due to this cooling and to the relative absence of convection carries much moisture down to the ground layers from the atmosphere. These two processes conduce to the process of condensation near the ground, and the formation of dew, especially if vegetation is present. Hence the low temperatures near the ground cause the space to have a smaller capacity for water vapor and also cause the removal under proper circumstances of much of the water vapor by condensation, producing a minimum of vapor pressure and absolute humidity near the ground just before dawn. This is the so-called *secondary minimum*.

"As the sun rises, it warms the ground and evaporates much moisture. The lapse rates at first are insufficient to cause much instability, hence the vapor pressure rises to the *primary maximum* occurring between 6 and 9 a.m. When the sun gets higher, convection becomes active, carrying much water vapor away from the ground layers. By the time the afternoon maximum temperature (dry-bulb) has been reached, the supply of surface ground water has been greatly depleted and the rate of evaporation from the ground has become less than the rate at which the ascending air currents and eddies carry the moisture aloft. Hence we have the *primary minimum* of vapor pressure (and absolute humidity) at the surface occurring about mid-afternoon in the summer at inland stations. The evening (*secondary*) *maximum* occurs as a result of the rapid subsidence of air at dusk or shortly thereafter when convection has greatly diminished, and also as a result of the comparatively small decline in temperature (dry-bulb) near the ground."

The characteristics of the curves in Fig. 5 check closely with Mr. Harrison's comments, although in some cases the evening peak is found at a later hour and occasionally there is no downward drop during the night. A representative example of this latter case may be noted in Fig. 6, on which the dry-bulb, wet-bulb and dew-point temperatures for each hour of the period of July 12 through July 16, 1934, for Dallas, Tex., have been plotted, and a smooth line run through the respective readings. The dry-bulb shows a very uniform daily cycle, reaching its peak in each day about 4 p.m. In general, the wet-bulb curve reaches its peaks and valleys ahead of the dry-bulb temperature, but has the same typical appearance. The dew-point, on the other hand, generally reaches its one daily peak around 8 o'clock in the morning, and its valley from 4 to 8 in the afternoon. Of greatest interest, however, is to observe, through the five days of record at Dallas just cited, a consistent climb of the daily peak dry-bulb temperatures, accompanying a steady daily decline of the dew-point and wet-bulb temperatures, showing once more the blanketing effect of increased moisture content of the air. While Dallas is somewhat unusual in failing to show a morning valley, the important characteristic common to most stations of a minimum dew-point during the afternoon is decidedly evident. Note also that the behavior of the dew-point, as illustrated in Fig. 6, is responsible for the peak of the Dallas wet-bulb temperature on Fig. 5 being

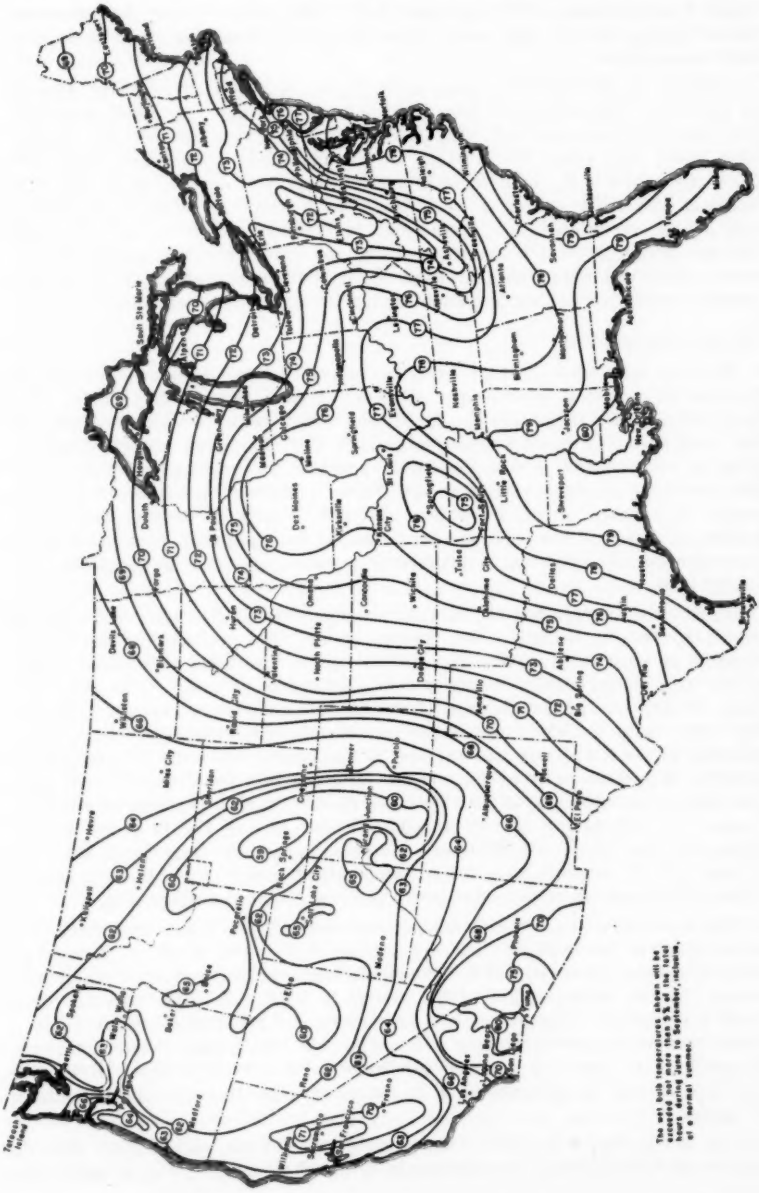
The wet bulb temperatures shown will be exceeded, on the average, during the three months June to September, inclusive, of a normal summer.

FIG. 7. SUMMER WET-BULB TEMPERATURE DATA

reached before noon, with a gradual drop taking place during the afternoon period, during which time many other locations experience their peak wet-bulb temperature.

Summing up the tendencies observed in the seasonal and daily curves, referred to previously, several major facts are obvious. Of these, the more important are: that the maximum daily temperature is attained as a rule for only a short time daily; that either the average or the maximum temperature is a poor basis upon which to build a judgment of a proper design value; that the seasonal and daily ranges vary through wide limits in different locations; that a wide variation in one factor may be accompanied by a relatively small one in another; that the maximum values of the three temperatures do not occur simultaneously; and that extreme dry-bulb temperatures *do not occur simultaneously* with either maximum dew-point or wet-bulb temperatures.

Design Temperatures

To meet the seasonal range, it is necessary that the equipment chosen be capable of operation with proper efficiency at the various daily loads. The daily temperature range and the speed with which changes take place call for the installation of control sensitive enough and equipment properly built to react to such control so that the daily variations are followed properly. The diversity between long ranges in one factor, combined with relatively short ranges in another, affects the choice of types of units to make up the entire system. Above all, however, the gross size of the equipment installed depends upon the outside operating temperatures assumed in the original calculations of the load.

Neither the average nor the maximum temperature is a satisfactory value on which to base a design. The former results in equipment of insufficient capacity to meet the load for over 50 per cent of the time. The latter would result in equipment which would never be loaded to capacity. The maximum value of any one factor does not exist simultaneously with extreme values in the other variables which also enter into the design calculation. Some compromise between maxima and average is the logical answer. An analysis of practice in various parts of the country indicates that installations based upon wet-bulb temperatures which are exceeded no more than 5 per cent of a normal season have given satisfactory results, without raising the first cost of the equipment too high. For instance, common practice in New York has been to use a 75 F wet-bulb as a design temperature, which, from the analysis, is found to be equalled or exceeded just 5 per cent of the normal season.

The hours that the wet-bulb temperature exceeds the 5 per cent value are, since close to the maxima, not consecutive hours, but occur in periods of relatively short duration, and with dry-bulb temperatures well below the maximum. The fly wheel action of the system is usually sufficient to carry over such a condition. Discomfort can be endured or production curtailed for a short time without experiencing a financial loss that would dictate additional expenditure to meet the extreme conditions with perfect results. Reference to Figs. 5 and 6 substantiates this statement regarding the short duration of peaks.

The dry-bulb and wet-bulb temperatures are not the only factors that do not reach their maxima simultaneously with other values. For instance, the

maximum dry-bulb temperature occurs after the peak of solar radiation has passed. The maximum dew-point may be experienced coincidentally with high wet-bulb temperatures, but it cannot exist simultaneously with the maximum dry-bulb temperature. While the maximum daily normal wind velocities occur in mid-afternoon, extraordinarily high wind pressures causing excessive infiltration of outside air are not experienced during high dry-bulb temperature periods. An analysis of Central Park Records shows that the dry-bulb and wet-bulb temperatures in New York reach or exceed their respective design values, which are well below the maxima, simultaneously only 37 hours in the normal year, scattered over eight different periods.

Considering the successful use of such values in past practice, the map shown in Fig. 7 has been developed to show the wet-bulb temperatures which are exceeded but 5 per cent of the time during a normal season. Similarly, Fig. 8 shows a map of dew-point temperatures which will be exceeded 5 per cent of the time during a normal season. The majority of the readings exceeding the wet-bulb values on the map occur during the 12 hours in the middle of the day. A much smaller portion of the excessive dew-point values occurs during that period, as was brought out in the discussion of Figs. 5 and 6. In fact, most of the high dew-point readings occur before 10 a.m. or after 7 p.m. Therefore, if a value of 5 per cent is satisfactory for a wet-bulb temperature design basis, certainly the same percentage is amply satisfactory on the non-coincidental dew-point temperature.

The peak dry-bulb temperatures all occur in the 12-hour span of the middle of the day. These points, combined with the fact that the daily range in dry-bulb temperature is much greater than either of the other two factors, dictate that the basis of choice of a dry-bulb temperature for use in design should be one that is exceeded a smaller percentage of the total time, or perhaps the same percentage of the daytime hours only. A choice of the correct dry-bulb temperature as that temperature which is exceeded but 5 per cent of the 12 hours of the middle of the day is a logical conclusion. Since all of the higher dry-bulb temperatures in the 24 hours occur during these hours, this is equivalent to saying that a satisfactory dry-bulb temperature is one exceeded no more than 2½ per cent of the hours in a normal season. The map in Fig. 9 is predicated upon the temperatures shown being exceeded no more than 2½ per cent of the time in a normal season.

Since high wet-bulb, dew-point and dry-bulb values do not occur simultaneously, it is logical that a coincidental dew-point calculated from the map values of dry-bulb temperature and wet-bulb temperature will be much lower than the map dew-point value. Similarly, the co-incidental dry-bulb temperature calculated from the value of dew-point and wet-bulb temperatures taken from the maps would be much lower than the dry-bulb temperature shown. Simultaneous use, therefore, cannot be made of the three map values in design, but they must be used in pairs in those parts of the design where the pair chosen have more effect than the third factor. In many problems a value in design is dependent on only one of these variables.

In some few special cases, design values approximating the peak temperature to be expected in one factor in a normal year must be used, in which case the values from Table 2 may be used as reasonably safe maxima. These are not the maximum readings known at each station, as, for instance, the normal

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TABLE 2—REASONABLE MAXIMUM TEMPERATURES AND ABSOLUTE MAXIMUM WIND VELOCITIES IN THE UNITED STATES

STATE	CITY	REASONABLE MAXIMA, TEMP. F			MAXIMUM RECORDED WIND VELOCITY MPH
		Dry-bulb	Wet-bulb	Dew-point	
Alabama	Mobile	95	82	80	87
Arizona	Phoenix	113	78	74	40
Arkansas	Little Rock	103	83	78	49
California	Fresno	110	75	66	41
	Laguna Beach	82	70	68	..
	Oakland	94	68	67	50
	San Diego	88	74	72	43
	Williams	110	80	71	..
Colorado	Denver	99	68	64	53
	Grand Junction	102	68	63	..
Connecticut	Hartford	...	82	..	58
District of Columbia	Washington	99	84	80	55
Florida	Jacksonville	99	82	79	58
	Miami	92	81	79	87
Georgia	Atlanta	101	82	79	51
Idaho	Boise	109	71	65	43
Illinois	Chicago	104	80	77	65
	Moline	103	83	80	..
Indiana	Evansville	102	82	79	60
Kansas	Wichita	110	79	74	68
Louisiana	New Orleans	95	83	81	66
	Shreveport	102	83	79	50
Massachusetts	Boston	96	78	76	60
Michigan	Detroit	101	79	76	67
Minnesota	St. Paul	103	79	75	78
Mississippi	Jackson	103	82	80	49
Missouri	Kansas City	109	79	76	57
	Kirksville	108	82	79	..
	St. Louis	108	81	76	91
	Springfield	98	79	76	52
Montana	Helena	97	70	60	54
Nebraska	North Platte	104	76	71	73
	Omaha	108	80	75	53
Nevada	Elko	101	64	58	..
	Reno	102	66	56	46
New Jersey	Camden	102	82	78	68
	Newark	99	81	77	..
New Mexico	Albuquerque	98	68	66	63
New York	Albany	97	78	75	59
	Buffalo	93	77	74	73
	New York	100	81	78	73
North Carolina	Raleigh	98	82	80	45
	Wilmington	94	81	79	53
North Dakota	Fargo	105	..	..	58
Ohio	Cincinnati	106	81	78	54
	Cleveland	101	79	76	60
Oklahoma	Tulsa	106	79	77	..
Oregon	Portland	99	70	68	43
Pennsylvania	Bellefonte	96	78	75	..
	Pittsburgh	98	79	74	56
South Carolina	Charleston	98	82	80	81
South Dakota	Huron	106	76	74	56
	Rapid City	103	71	66	..
Tennessee	Knoxville	100	79	76	59
	Memphis	103	83	80	58
Texas	Amarillo	101	75	70	65
	Brownsville	96	80	79	80
	Dallas	105	80	76	63
	El Paso	101	72	69	60
	Houston	100	81	79	63
	San Antonio	102	83	82	56
Utah	Modena	97	66	61	..
	Salt Lake City	102	68	64	53
Washington	Seattle	86	70	67	59
	Spokane	106	68	58	41
Wisconsin	Green Bay	99	79	75	53
	La Crosse	100	83	81	69
Wyoming	Rock Springs	91	62	58	..

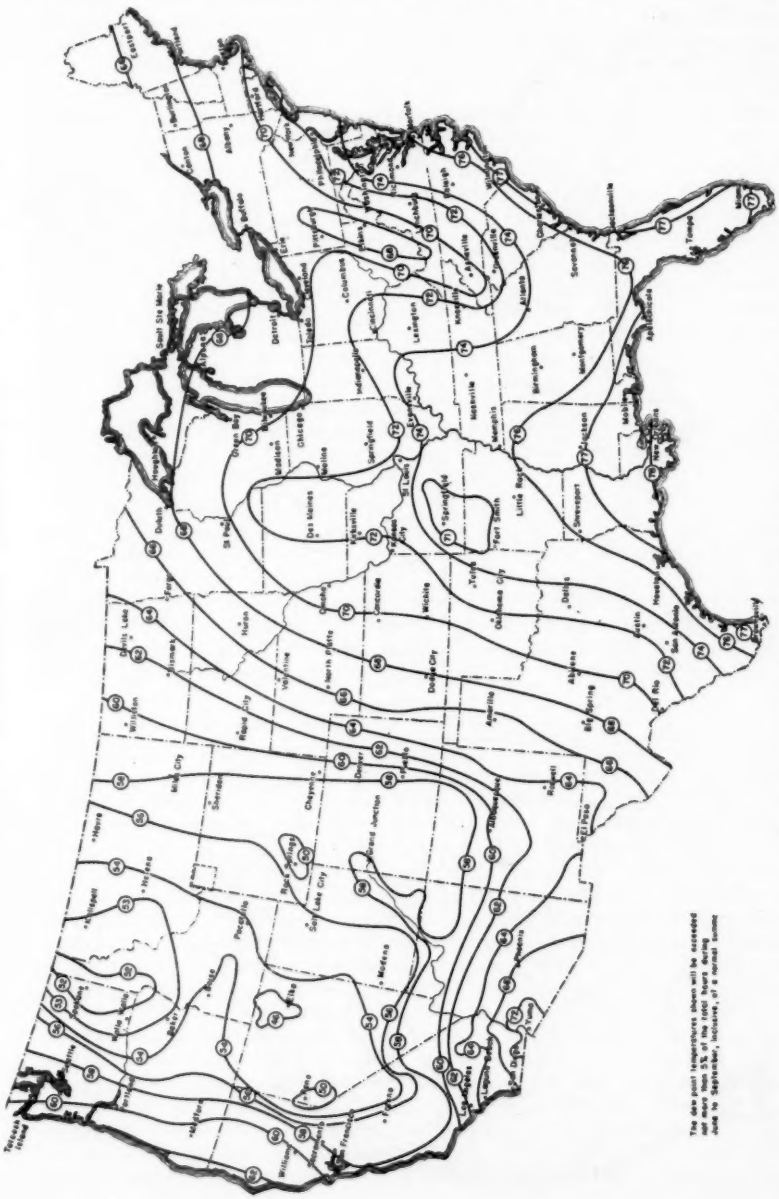
The dew point temperatures shown will be exceeded
on the average 10% of the hours during the period
June to September, inclusive, of a normal Summer

FIG. 8. SUMMER DEW-POINT TEMPERATURE DATA

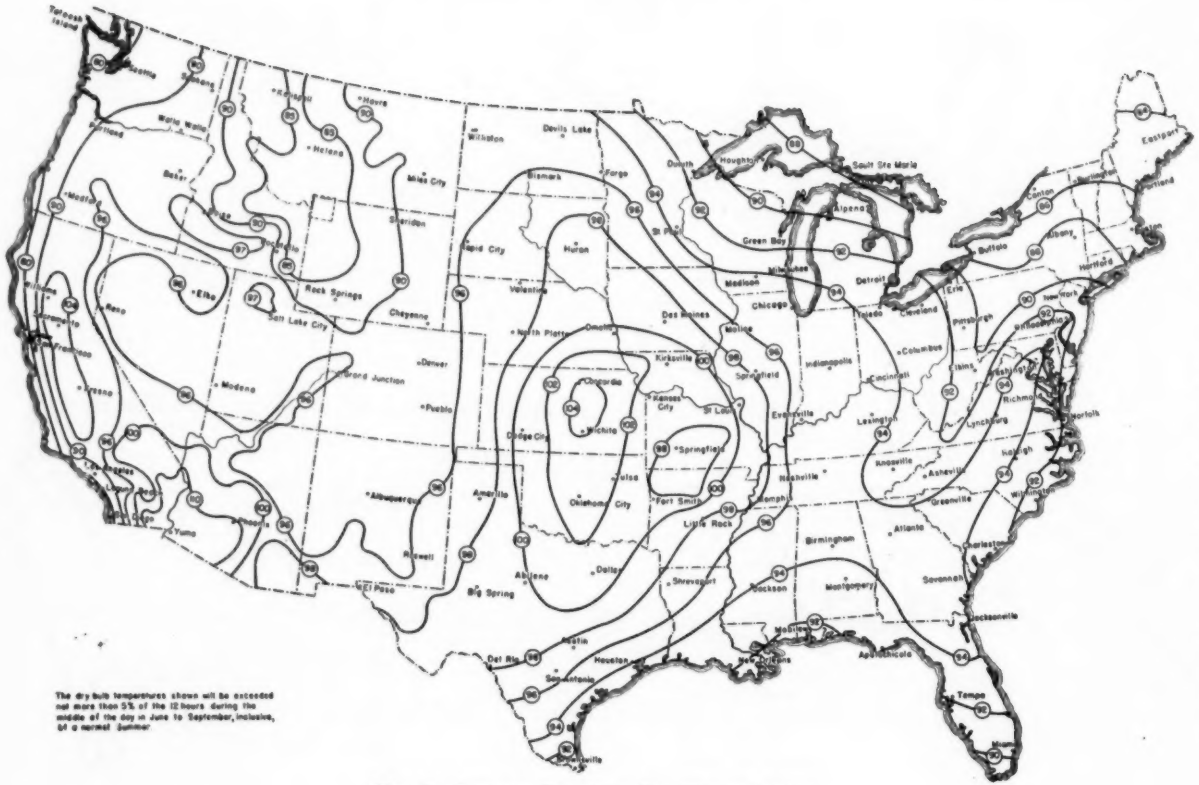
The dry-bulb temperatures shown will be exceeded not more than 5% of the 12 hours during the middle of the day in June to September, inclusive, of a normal Summer

FIG. 9. SUMMER DRY-BULB TEMPERATURE DATA

maximum wet-bulb temperature in New York is 81 F, yet a perfectly authentic reading of 88 F was recorded at 3 p.m. on July 28, 1931. This was a freak reading, lasting only a few moments, then dropping back below 80 F. Most exceptionally high readings in any of the variables are due to freak and abnormal conditions, and since sound engineering judgment would dictate ignoring such data, the table disregards values not normally reached each year.

When a combination of freakish weather circumstances occurs, an abnormal day will result, upsetting all expectancy, the effect of which cannot be counteracted in economical design, but must simply be endured because it is too expensive to do otherwise. For instance, on two occasions only during the period of 1925 to 1938, inclusive, the wet-bulb temperature at the Central Park Station of the City of New York reached and maintained an average reading of 80 F or more for a period of over seven consecutive hours. These took place on August 1, 1933, and July 9, 1936, and a graph of the wet-bulb and dry-bulb temperatures for these days is shown on Fig. 10. Later, in the same month of July, 1933, for the five days from the 20th to the 24th, inclusive, the wet-bulb remained steadily between the limits of 70 and 78 F during 8 a.m. to 5 p.m. each day. It is logical to design equipment to carry through the latter condition, but it is not warranted to meet in design the former condition, inasmuch as it occurred but twice in 14 years.

Throughout this paper a great deal of discussion has referred to the *normal* year, or the *normal* day. Actually, there is no such thing as a normal day, week, month or year in weather data, as the only thing consistent throughout about the weather is its inconsistency. Conditions in each of the factors are changing continuously, and the permutations are so great that for the conditions to duplicate any predicted combination over an appreciable period of time would be beyond the probabilities. One variable alone may follow the expectancy exactly, but it is beyond reason to expect all of them to do so simultaneously.

Influence of Geophysical Features on Temperatures

In each of the three maps (Figs. 7, 8 and 9) may be noted, reflected on a broad scale, the effect of the geophysical characteristics which have been touched upon at various points throughout the discussion. The wet-bulb and dew-point values near the sea coast are higher, and the dry-bulb temperatures lower, due to the influence of the ocean, which decidedly affects both the moisture and sensible heat content of the air. The general effect of elevation may be seen in the downward dips of all temperature lines around the Eastern mountain barrier; by the steady fall of all temperatures between the Mississippi Valley and the Rocky Mountains; and even the relatively small elevated plateau of the Ozarks in Missouri and Arkansas reflects in reduced values compared to surrounding territory. In this latter case there seems to be a shielding effect on the land to the northwest, possibly due to the fact that prevailing winds to these regions pass over the Ozarks en route, as is evidenced by the bulges in the wet-bulb and dew-points lines around Kansas City, From the Rocky Mountains east to the coast, the variations in elevation are gradual enough to depend upon values interpolated between adjacent temperature lines, but west of a north and south line drawn through the eastern border of New Mexico, the country is so broken that the temperature lines can be taken only as a general indicator of conditions. To be accurate and thorough, every mountain

and valley should be enclosed by a series of closed isothermals, with the minimum temperatures at the peaks of the mountains and the maximum at the center of the valleys. Certain major conformations reflect on all the maps, as the high ridge from Montana down through Wyoming to Colorado; the coast range in Western California; and the low areas in Central Washington, Central California, the Colorado River Valley and Great Salt Lake. The Climatological and the Aerological stations are normally located at cities and cities are built in valleys, therefore, the map values in broken country are for valley conditions. To determine completely temperature expectancy in moun-

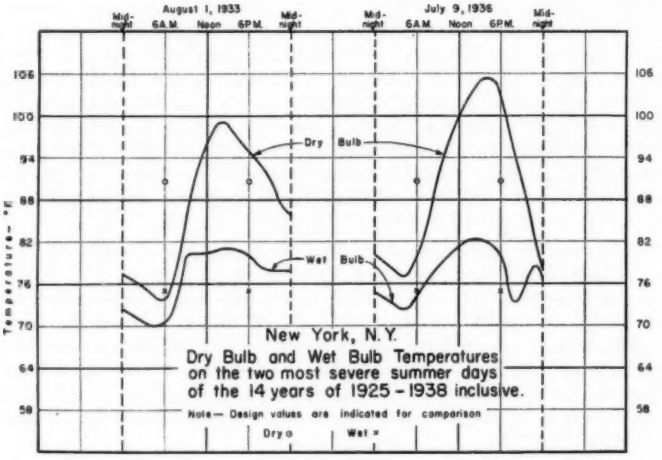

FIG. 10. EXTREME TEMPERATURE CONDITIONS IN NEW YORK, N. Y.

tainous country would require data from several times as many stations in such country as are needed for the same area in plains regions, whereas the present availability is the reverse.

WIND DATA

Wind Movement During the Summer Period

While there is a definite trend of winds in the various sections of the country more or less characteristic of the season, the proportion of time the wind blows from the preponderant direction is not as great as is commonly thought. On the map in Fig. 11, the average hours that the wind blows from each direction in July have been portrayed at 71 locations throughout the United States. Here again the geophysical influence is directly noted. Starting on the East Coast, the preponderant direction is from the Southwest, in the Northern part to Southeast in Florida, breaking into indeterminate expectancy over the Eastern mountain range, and showing a definite tendency to winds from the South

in the Mississippi Valley. The broken country through the Rockies shifts the preponderant winds at right angles at some locations relatively close together, as, for instance, the group of six cities whose graphs are intercepted by a line from San Francisco to Havre, Mont. In this group Boise and Fresno lie at relatively low elevations, and Winnemucca and Helena at considerable heights. It is of interest to note that a map of this same character covering the annual wind movement shows the preponderance of movement over the 12 months to be in the same direction as for July for approximately 90 per cent of the locations, although in each case the percentage of total time that this direction exists is less on the annual basis.

With reference to wind velocities, for which the July average value in miles per hour is shown in the circle at each location on the map and the maximum recorded velocity for the entire 12 months is included in Table 2, considerable judgment and understanding of the individual location must be exercised in adopting any figure in the design of equipment. Wind values, perhaps more than any other variable, are decidedly affected by the location of the observation. Velocities near the ground are normally lower than those at higher elevations and the character of surrounding obstructions has a great deal to do with the readings. A location anywhere in the center of a large city is affected by the mountainous character of the surrounding objects which obstruct air movement, generate convection currents and cause mechanical turbulence. Neither the direction nor the velocity taken at another location can be accepted without question as applying to the observed point. If taken on the top of a building, the observations are being made on strata of higher velocity than normally exist near the ground. For instance, the records of the Climatological Station instrument, located on top of the Whitehall Building in New York, at an elevation of 304 ft above street level, show a July average over the 20 years of record of 12.6 mph. The Central Park Observatory of the City of New York, on the other hand, whose anemometer is 62 ft above the ground, shows an average of but 8.7 mph during July for a period of 17 years.

For those designs affected by the maximum wind force, such as the strength of building construction, a wind velocity well above the maximum recorded value should be used, this intensity frequently being set by building codes. For those designs where the reasonable expectation of minimum velocity dictates, values well below indicated averages should be chosen, both because the reference data are taken at considerable elevations and since every location experiences periods of calm, or zero wind velocity, especially in the summer months. It is recommended that values not over a half of the average velocity values shown on the map be chosen for the design of such installations, as the wind intensity will be even less than this chosen value for from 20 to 50 per cent of the time, depending upon the location, and especially so in those locations having low averages.

SUNSHINE DATA

Sunshine Distribution Throughout the United States

The use of solar radiation data in design cannot, in the present status of data available, be used as accurately as those covering the other factors. This is due to lack of widespread data on the intensity of the sun's rays, coupled

FIG. 11. SUMMER WIND DATA

with incomplete research on the application of such data, if they existed. In only a few locations throughout the United States are any records taken of solar radiation. Even though such records existed from all of the 200 Climatological Stations, for instance, it would be impossible to prepare a reliable comparative map containing solar radiation values safe for design purposes, similar to Figs. 7, 8 and 9 on temperatures. Each of the records would fit the peculiar characteristics of that one location only, and a very few miles away would find different values, as sunshine intensity is most seriously affected by entirely local mechanical characteristics. It is theoretically a constant at all points on the same elevation and latitude, for the same hour and day of the same individual year. Practically that fixed intensity is reduced by the amount of moisture, dust and smoke in the air. Thus, a measurement of the sun's intensity at one location becomes, if compared to the theoretically possible intensity, purely a measure of relative clarity of the local atmosphere. The dry-bulb temperature, to a certain extent, reflects this value and is at the present time the only general comparative measure, although a poor one, of the sun's intensity.

A certain usage may be made, for comparative purposes, of the average hours of sunshine per day, month or season. Comparison between the actual hours experienced to those theoretically possible, permits an approximation of the total sun load per season through the use of any one of a number of tables available showing the heat gains at times of full sunshine for various latitudes on surfaces of different exposures, textures, colors, etc. The map in Fig. 12 illustrates the average actual daily hours of sunshine in the season of June through September with lines of maximum possible sunshine shown for comparative purposes.

Again the effect of the geophysical characteristics may be observed, although not with the predictable result as in the case of temperatures. The mountain ridges in the East and through Wyoming have a reduced number of hours of sunshine, compared to surrounding territory. This is due to the broken character of the country, rather than the elevation, since the general trend of duration of sunshine per day increases from the Mississippi to a high point in California. The West Coast has greatly reduced sunshine compared to the mountainous region a short distance inland, which is the exact reverse of the situation on the East Coast.

SUMMARY

Except for the years of Aerological data utilized here, there are no data available in usable summarized form in the records of the Weather Bureau that will permit a uniform investigation of temperatures over the full 24-hour daily period. The four years used were slightly higher in dry-bulb temperature on the average than the mean of the maximum and minmum temperatures for all of the years of record at the corresponding Climatological Stations, but the differentials were not great enough to do more than render the final results conservative. In those locations where additional years of data were available, comparisons were made to the data developed, and it was found that doubling the period of analysis made no practical change in the results.

The preceding survey is a report of the present status of research extending

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FIG. 12. SUMMER SUNSHINE DATA

over the past eight years. An earlier product of this research appears as design temperature recommendations in THE GUIDE of the Society. These present published data in THE GUIDE are based on the only data available at the time they were developed, consisting of the regularly published data as contained in Figs. 1 and 2, and average maximum dry-bulb data as accumulated by the Climatological Division, coupled with a computed depression of the wet-bulb temperature. That these data differ from those of this latest analysis is further proof of the danger of inaccuracy resulting from the use of average or maximum data in the developing of weather design data.

Further study of solar radiation should be made on a national scope. While information on this subject could not be utilized practically at the present time, it will become increasingly important, and by the time sufficient data can be accumulated, proper use can be made of the results. Additional studies of wind movement are being made at present by the Weather Bureau, and in view of the relatively minor use of data on wind in the heat exchange industry, more detailed investigation seems unnecessary at the moment. It is very much to be desired that wet-bulb temperatures over the entire 24-hour period be taken and made available to the public by the stations of either the Climate and Crop Weather or the Aerological Divisions of the Weather Bureau, as such readings are necessary for comparison in detailed studies of local conditions.

These local researches are needed to answer the questions as to the temperature differentials existing between different points in the same city. Undoubtedly, spot temperature readings at street level in larger cities have neither the same simultaneous values nor do they follow the exact daily cycle of the reference readings at the Weather Bureau. It is believed, however, from a comparison of Climatological and Aerological data, that the average daily temperatures will not be affected greatly by such differences in location, but the daily peaks and valleys of the temperature curves will be both foreshortened and delayed by the heat absorption of the masses of masonry, concrete and steel in the center of a city. Studies of this nature to date have been of a sporadic nature, usually incomplete, sometimes inaccurate, and frequently have done more harm than good, due to the misleading character of the resulting data. Such investigations, conducted in a uniform, accurate manner throughout the country by interested local organizations would be of greatest value in most accurately adjusting the design conditions to fit the operating requirements of the individual problem.

CONCLUSIONS

The development of satisfactory data for design requires readings taken at uniform intervals throughout the 24-hour daily period. Neither maximum readings nor average readings are sufficient to establish a basis for design. Maxima in the various factors do not exist simultaneously. The data presented are a sufficiently complete forecast of the normal expectancy in weather performance to be used in design at all points East of the Rocky Mountains, and are a reasonably correct guide for Western valley locations. More detailed information regarding individual localities should be collected by local interests, but such local effort should be coordinated throughout the country. Addi-

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tional investigation of weather expectancy of value to the heat exchange industry throughout the country will be possible only through more complete reports of the Weather Bureau than are at present available.

ACKNOWLEDGMENTS

The accumulation of unpublished Aerological data was made possible through the cooperation and assistance of D. M. Little, chief of the Aerological Division, and P. A. Miller, also of the Aerological Division. L. P. Harrison assisted with data and advice regarding the behavior of weather phenomena. J. B. Kincer, chief of the Climate and Crop Weather Division, gave suggestions and provided office space for use during the collection of information from the Climatological Division. W. W. Reed of the Climatological Section furnished data on the type of wind analysis presented and directed the search of the Climatological temperature files.

The data included in the analysis are based upon the program of necessary factors adopted by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS' Technical Advisory Committee on Weather Design Conditions. Original records from the United States Weather Bureau were obtained and analyzed by the author in cooperation with the Society's committee composed of T. H. Urdahl, chairman; J. C. Albright, E. W. Goodwin, J. B. Kincer, J. W. O'Neill, John Everetts, Jr., A. C. Grant, O. A. Kinzer and L. S. Ourusoff.

DISCUSSION

W. L. FLEISHER: I want to express to Mr. Albright, and also to explain to the Society at large, that this work is an astounding piece of work. It has been, in a way, a life interest of Mr. Albright, and he has devoted not only an enormous amount of time and thought, but also a great deal of money to the development of the charts and the analysis of them, which he has presented so admirably to all of us. Very generously he has cooperated with the research activities of this Society in making available the work that he has done. All of us who design or are connected in any way with the design of air conditioning for summer owe Mr. Albright a great debt.

J. E. RYAN:[1] I wonder if it would be possible to effect a constant for any particular locality by which the solar intensity, the theoretical solar intensity measurement, might be multiplied, in order to get a decent design factor?

In general, if we have an equal cloudiness throughout a day, the curve should be a certain fraction of the amplitude of those which I showed in a previous paper.[2] At least, a design condition would be similar in shape to the theoretical, and, if it were possible to take solar intensity measurements over a long period of time at each locality, could we not find a factor to apply in each case?

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[1] Engineering General Dept., General Electric Co., Schenectady, N. Y.

[2] Effect of Heat Storage and Variation in Outdoor Temperature and Solar Intensity on Heat Transfer Through Walls, by J. S. Alford, J. E. Ryan and F. O. Urban. (ASHVE TRANSACTIONS, Vol. 45, 1939, p. 369.)

G. L. WIGGS: I think this paper is an admirable contribution to the knowledge of temperatures in the United States. Being a Canadian, I regret to see that all these beautiful lines stop at that invisible line of the border. I, in my small way, have tried to work up some of this information for some of the Canadian cities, and even if your stations here have not done a particularly good job, you should see the Canadian record. We have, of course, records there that have extended back longer, I think, than any meteorological station here, and I have toyed with some of those figures. These are actual summer cooling data, but one of the things that has interested me is that, having nothing to do a few years back, I decided to study the degree-days, and I took the McGill record for 68 years and worked out the degree-days daily, and plotted them on a piece of chart paper. It was interesting to see that those went up and down, but the thing which was of greater interest was that the temperatures were slowly rising in Canada with diminishing degree-days. That would indicate that Mr. Albright might have to go through this calculation about once every ten years to bring it up to date, but I think it is an admirable contribution and I hope that the Society will continue with this, particularly to bring these lines up to at least the southern part of Canada.

T. H. URDAHL: There are two topics which perhaps form the basis of more conversation in this world than any other. One is the weather and the other is health, and health is rather dependent upon weather. Mr. Albright's paper not only presents exceedingly valuable basic data, but more especially it points the way in methods of analysis to continue an investigation of quantitative weather. We all know pretty much what kind of weather we have in different localities, but a very important question is: How much of each component that makes up the weather exists during a given period of time?

MR. FLEISHER: During a recent trip for the Society in connection with research, I discovered in the Middle West, or further west, toward the Rockies, and in all those sections of the country in which our new irrigation systems had been installed, that the wet-bulb conditions had definitely changed. There is a large territory in the United States in which formerly, due to low wet-bulb conditions, they had been able to use evaporative cooling. Since our new irrigation scheme has been finished and consummated, a very large section of that part of the country has now found it ineffective. I would ask the author of the paper whether he was cognizant of that or whether he has investigated the changes or whether it is simply local gossip.

J. C. ALBRIGHT: With reference to Mr. Ryan's comment on solar radiation, it is possible to get solar radiation data at individual points but such data would be affected by the present industrial condition at such points. Air clarity changes with variations of smoke and dust concentration and thus it is questionable as to whether, even with a reasonable number of points, we could possibly design a map of solar radiation values in the United States on national scope that would be satisfactory for general use.

It seems to me that this sort of research is entirely a subject for local survey, financed by local organizations for local use only. It is hoped that some of this work will be done in the next few years. If enough of such data were obtained a national map of some value could be made, but at the present time, it is entirely impractical.

With regard to Canada, nothing would be more pleasing than to have good information for Canada. If the proper kind of data were obtained, which would be a monthly record in some convenient form of hourly dry-bulb, wet-bulb and dew-point temperatures, the maps could be extended readily and in addition it would help us to induce the United States Weather Bureau to do likewise.

So far we have not been able to get just what we need for our industry in the

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United States. Washington is interested in forecasting for the general information of the public, for agriculture, and other uses which have been basic with the department since its inception. Aviation, of course, is very admirably served at the moment with weather data. Industry ought to get busy and obtain similar service for their purposes.

Regarding Mr. Fleisher's comments on irrigation, there has been a considerable amount of argument pro and con on that particular subject. For instance, in the Imperial Valley, Brawley, Calif., there are perfectly authentic data showing 90 deg wet-bulb temperatures and an average of over 80 deg wet-bulb temperature for an entire month.

As to whether normal irrigation, excepting for two or three very isolated spots, particularly affects the wet-bulb temperature, such investigation as we have conducted does not seem to show a very great variation. Around Phoenix there is a small amount of variation, and quite a bit in the Imperial Valley, but the stories obtained from local sources are exaggerated, and a small change is magnified in the telling to be a considerable factor. Certainly it is changing the conditions, but not quite as much as local color would have one believe.

No. 1125

APPLICATION OF SUMMER WEATHER DATA IN DESIGN

By JOHN EVERETTS, JR.* (*MEMBER*), NEW YORK, N. Y.

THE method of determining weather design data for the four summer months to be used in air conditioning and other heat exchange problems where the variations of atmospheric conditions are important has been covered in a previous paper.[1] It is the purpose of this paper to demonstrate the practical application of these data in air conditioning.

An analysis will first be made to segregate the various components of the load on an air conditioning system and to identify the various portions of the data given in the paper referred to with the corresponding sections of the design. To demonstrate the actual use of the data in design, a typical air conditioning problem will be assumed covering four representative cities and the calculated loads resulting from the use of the weather design data will be analyzed and compared.

The only function of an air conditioning system is to maintain a specified atmospheric condition within an enclosure irrespective of the existing conditions outside the enclosure. Aside from the fixed loads of people, lights and motors, the design of an air conditioning system and equipment is dependent entirely on the variations and intensities of the dry-bulb, wet-bulb and dew-point temperatures, solar radiation, and wind effect, and it is therefore necessary that these weather factors be properly chosen so that the system and equipment as designed will do its specified duty and still be within the limits of economic consideration. Heretofore, the choice of design conditions has been made through long years of experience, rough perusal of Weather Bureau reports where these reports were available, and trial and error methods. These methods led to a considerable variation in specified design conditions and when used in the calculation of a system, rendered that system either inadequate to meet the demands which existed or made it so large as to be economically unsound for the particular duty. For example; in New York conditions have been specified ranging from

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* Air Conditioning Engineer.

[1] Analysis of Summer Weather Data in the United States, by J. C. Albright. (ASHVE TRANSACTIONS, Vol. 45, 1939, p. 397.)

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90 to 98 F for dry-bulb and from 72 to 80 F wet-bulb, with no consideration for the dew-point. The design temperatures for New York as shown in the paper previously cited are 90.7 F dry-bulb, 75.0 F wet-bulb, and 72.4 dew-point; each temperature having been chosen by combining the analysis of Weather Bureau data and the practical requirements of an air conditioning system.

The treatment of atmospheric air is made complex by the fact that it is a mechanical mixture of dry air and water vapor involving sensible heat, latent heat and total heat, each of which is a function of either dry-bulb, wet-bulb, dew-point, or various combinations of these temperatures.

These interrelated functions are further complicated by the fact that while the increase in sensible heat is proportional to the increase in dry-buib temperature, the increments of latent heat are non-uniform for equal temperature ranges in different parts of the wet-bulb scale.

LOAD DETERMINATIONS

It has been shown that the dry-bulb, wet-bulb, dew-point, sunshine, and wind movement vary considerably over the country and vary independently of each other, with no co-incidental relationship. These weather factors affect an air conditioning system in two ways, (1) total load, and (2) apparatus design. To analyze the total load, it must be divided into its component parts and each part separately considered with the weather factor by which it is affected. The component parts of the total load for the purposes of such analysis are (1) transmission, (2) ventilation, (3) solar radiation, and (4) internal load.

Transmission Load

The transmission load of an air conditioning system is purely a sensible heat load and is dependent upon variations in dry-bulb temperature and wind movement. The wind movement is only a minor consideration in this connection inasmuch as it has been taken into account in the preparation of the tables showing heat transfer factors for the particular walls and glass enclosing the conditioned space. The maximum daily wind velocity occurs at the time of maximum dry-bulb temperature but very seldom exceeds the 15 mph upon which the heat transfer factors are based in the tables and, therefore, may be neglected in the transmission load but must be considered in the ventilation load as infiltration.

The only other variable is the dry-bulb temperature, but, as small changes in this factor have a major effect on the total transmission load, it is necessary that the choice of a proper outside dry-bulb temperature be made with much greater care than needed in the consideration of wind. This selected temperature, hereafter referred to as the *design dry-bulb* is given in Table 1 for the four cities selected for analysis.

With a constant maintained condition in a conditioned space the transmission load will vary directly with the dry-bulb temperature. As the design

dry-bulb for Kansas City is higher than that of New York, the transmission load for a given space will be higher for Kansas City than for New York in proportion to the difference between the design dry-bulb and the maintained dry-bulb. While the absolute value of the transmission load in tons is proportional to the dry-bulb differential, the percentage of transmission load to the total calculated load is not proportional to the dry-bulb differential because the total load is affected by the other fixed and variable components of the total loads.

The transmission load based on maximum dry-bulb temperatures apparently indicates that in some cases the load will be double or more that calculated on the basis of the design dry-bulb. This is particularly true of cities having a

TABLE 1—TEMPERATURE DATA[a]

		NEW YORK	PITTSBURGH	KANSAS CITY	NEW ORLEANS
Design Temperature	D.B.	90.7	91.2	101.5	92.3
	W.B.	75.0	72.6	75.5	80.1
	D.P.	72.4	69.7	71.5	77.9
Noon Avg	D.B.	75.8	77.8	83.7	86.1
	W.B.	65.7	65.3	67.9	74.8
	D.P.	59.6	57.7	59.2	70.1
Normal Max.	D.B.	100.0	98.0	109.0	95.0
	W.B.	81.0	79.0	79.0	83.0
	D.P.	78.0	74.0	76.0	81.0
9. A.M. 5 P.M. Avg	D.B.	78.8	77.8	84.1	85.5
	W.B.	67.2	64.7	68.4	76.5
	D.P.	61.2	57.3	60.2	73.0

[a] All temperatures are in degrees Fahrenheit and refer to the four summer months.

wide range of daily and seasonal temperature, such as New York. These calculated maximum loads are not real demands upon the air conditioning system as these peak temperatures occur so infrequently and are of such short duration that the flywheel effect due to the heat lag in the wall will prevent the extreme temperatures from causing an exorbitant overrun of maintained temperature. Spaces having large window areas will be affected more quickly by extreme conditions since there is practically no heat lag in glass and the effect will vary directly with the glass wall ratio. The total effect of this increased transmission load is also proportional to the ratio of the transmission load to the total load which will reduce the over-all effect in proportion to this ratio.

As the outside dry-bulb falls off, the transmission load will be reduced until the outside dry-bulb is equal to or slightly below the maintained dry-bulb. However, it is erroneous to assume any reduction in operating load due to the outside dry-bulb falling below the maintained condition because of the heat

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lag in the building and the desirability of having a lower inside temperature as the outdoor temperature falls.

Ventilation Load

The ventilation load is the load produced by the outside air brought into the conditioning system to provide ventilation for the occupants and to offset any possible infiltration into the conditioned space. Infiltration is a direct function of the wind velocity and should be calculated in accordance with THE GUIDE procedure to determine whether the infiltration or ventilation requirements are the greater. The larger amount should then be used as the quantity of outside air to be conditioned.

As the outside air contains both sensible and latent heat in varying ratios it is necessary to calculate the total heat removal on the basis of wet-bulb temperatures, since the total heat content of the air is constant for a given wet-bulb temperature regardless of the variation of the sensible-latent heat ratio. The total heat increase per degree in wet-bulb temperature is not constant but increases as the wet-bulb temperature increases. This fact is important in those localities where high wet-bulb temperatures are predominant, and in problems where the outside air quantity is appreciable. As variations in design wet-bulb mean greater changes in calculated load than equal changes in design dry-bulb, it is essential that the proper wet-bulb temperature be used in the calculation of the ventilation load. These *design wet-bulbs* as set forth for the four reference cities in Table 1, indicate the wide variations encountered in different parts of the country.

TABLE 2—COINCIDENT TEMPERATURES[a]

	NEW YORK	PITTSBURGH	KANSAS CITY	NEW ORLEANS
D.B. Coincident for Design W.B. and Design D.P....	81.5	79.0	85.5	86.5
D.P. Coincident for Design D.B. and Design W.B.....	68.5	64.0	63.0	75.5

[a] All temperatures are in degrees Fahrenheit and refer to the four summer months.

Ventilation loads based on maximum wet-bulbs will have more of an effect on the total load than that of excess transmission. First, the effect of wet-bulbs higher than the design have an immediate effect as there is no lag in the effect of the wet-bulb on the system. Secondly, the ventilation load is a much greater part of the total load than any other individual item with the exception of certain industrial problems, and third, as the wet-bulb increases the total heat per degree increases to give an accumulative effect to the overload condition.

The compensating features which reduce the value of this overload are (1) the infrequency of occurrence of excessive wet-bulb temperatures, (2) the short duration of their existence, (3) the available overload capacity of a system, and (4) the fact that the maximum wet-bulb does not generally occur at the same time as the maximum dry-bulb, consequently the transmission load is less than the designed value which will offset part of the ventilation overload.

The ventilation load, contrary to the transmission load, can be assumed to reduce the operating load as the wet-bulb temperature falls. If a suitable air washer is used, it is possible and quite practical to do evaporative cooling whenever the outdoor wet-bulb is at or below the required delivered dew-point.

Solar Radiation

The load caused by solar radiation is entirely sensible heat and is considered either as additional heat transmission or as a separate sensible heat load depending upon the estimator's choice of method by which this load may be calculated. Since the data available and the method of calculation of this load are at present of such nebulous nature, discussion of them is omitted. As the solar radiation is a small proportion of the total load in the problem chosen it is considered best to include it as part of the internal load as a constant throughout the comparative calculations.

Internal Load

The internal load consists of both sensible and latent heat produced by people, lights, motors, and other miscellaneous heat producing sources. Since this load is not affected by any of the weather factors it may be considered as a constant in comparing similar problems in different locations.

APPARATUS DESIGN

The choice of air conditioning apparatus is definitely influenced by the dry-bulb, wet-bulb and dew-point temperatures, depending upon the type of system selected.

An air washer system is affected only by the wet-bulb temperature and. therefore, need only be designed for a total heat removal from entering to leaving wet-bulb. As the total heat is constant for a given wet-bulb temperature, the dry-bulb and dew-point are unimportant in the design of this type of system and the wet-bulb differential to be maintained determines the operating characteristics.

A chemical dehumidifying system is affected by both the wet-bulb and the dew-point, the wet-bulb indicating the total heat to be removed from the system and the dew-point indicating the total amount of moisture to be removed. The absorber capacity and regenerator design in a chemical dehumidifying system are fixed by the amount of moisture to be removed, which value is a function of the dew-point temperature.

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The dew-point existing at the time of simultaneous occurrence of the design dry-bulb and wet-bulb temperatures would be too low a value, and would result in the installation of inadequate chemical dehumidifying equipment. Instead, the design dew-point temperature should be used and considered to exist simultaneously with design wet-bulb temperatures for purposes of fixing the capacity of such apparatus. Referring to Table 2 where these design dew-points are shown for the four cities analyzed, it can be readily calculated that for Kansas City the moisture removal based on 71.5 dew-point would be more than three times that required for a dew-point of 63 F when reducing both to a required dew-point of 58 F. The same calculations for New Orleans show only an increase of 19 per cent, which can readily be understood by comparing the range of the dew-point temperatures of the two cities, as that for New

TABLE 3—DESIGN LOAD

	NEW YORK		PITTSBURGH		KANSAS CITY		NEW ORLEANS	
Load	Tons	Per Cent	Tons	Per Cent	Tons	Per Cent	Tons	Per Cent
Trans.	3.1	13.8	3.3	16.4	6.2	24.1	3.5	12.2
Vent.	8.2	37.0	5.7	28.5	8.5	33.1	14.4	49.8
Int. Load	10.9	49.2	10.9	55.3	10.9	42.8	10.9	38.0
Total	22.2	100.0	19.9	100.0	25.6	100.0	28.80	100.0
Ratio to Pittsburgh	1.11		1.00		1.29		1.45	

Orleans indicates small daily and seasonal changes, while that for Kansas City shows wide variations in daily and seasonal range.

Cooling coil capacities are affected by dry-bulb, wet-bulb, and dew-point temperatures and consequently, must be designed on the basis of these three temperatures being coincident. The dry-bulb temperature affects the mean effective temperature (M.E.D.) between the air and the coil surface; the total heat removal of the coil reflects the wet-bulb temperature; and the dew-point varies the transfer rate of the coil by changing the sensible-latent heat ratio. The capacity of the coil decreases with a decrease in dry-bulb and increase in dew-point temperature; hence, the design should be predicated upon the relatively low dry-bulb which occurs with design wet-bulb and design dew-point conditions. The dry-bulb temperatures coincident with the wet-bulb and dew-point design conditions are shown in Table 2 for the four cities indicated and may be compared with the design dry-bulbs as indicated in Table 1.

PROBLEM

The example used for comparison is that of a small store for which the maintained conditions in the occupied spaces are 80 F dry-bulb, 67 F wet-bulb,

and 60 F dew-point, and the delivered dew-point is 58 F. The load values given in Table 3 are calculated on the basis of the design temperatures in Table 1 and for operating hours from 9 a.m. to 5 p.m. six days per week during June to September inclusive. Pittsburgh has been taken as a base and the other cities have been compared with Pittsburgh to show how the results vary. The four cities chosen cover a considerable range in design, maximum, and average temperatures. It is of interest to note that while some locations have the same average temperature conditions, they do not have the same design values, and that other cities show higher design temperatures against lower noon averages. This is also true for maximum temperature comparisons and definitely indicates that maxima and averages are no criteria upon which to base the design of a system.

Design Load Analysis

The design load as shown in Table 3 is based on the design temperatures given in Table 1 for the four cities indicated, and the maintained conditions given in the problem.

It will be noted that the transmission load for New Orleans is higher than that of either New York or Pittsburgh, yet is the lowest in terms of percentage of total load. This is due entirely to the exceptionally high ventilation load which is the greatest of the four cities in both calculated load and total load. Pittsburgh shows the lowest ventilation load by virtue of its low design wet-bulb and also the lowest total load because of both its low design dry-bulb and wet-bulb.

The most important relationship in the design load analysis is the ratio of total loads to Pittsburgh. With similar jobs in Pittsburgh and New Orleans this analysis shows that the capacity of the system in New Orleans must be 45 per cent greater than that for the same system in Pittsburgh. This relationship is of particular importance to the manufacturers and engineers who recommend and specify a given sized unit for rooms of equal cubage and located in different parts of the country.

Maximum Load Analysis

The calculations of maximum loads given in Table 4 are based upon the normal maximum dry-bulb and wet-bulb temperatures given in Table 1, and calculated in the same manner as for the design loads. It has been shown in the previously mentioned paper that simultaneous occurrence of maximum dry-bulb and wet-bulb conditions is impossible. However, the effective transmission load equivalent to that of maximum temperature may occur at the same time as maximum wet-bulb temperatures, due to high heat capacity of walls and the building-up of high temperatures in these walls during such times as a week-end shut down period.

Kansas City has the highest transmission load of the four cities, while New York and New Orleans are equal in ventilation load. New York shows a

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greater total load than New Orleans which is an absurd conclusion as common sense would dictate and is further evidence that maximum temperatures should definitely not be used in the design of a system. The ratio of the total maximum load to that of Pittsburgh shows surprisingly little variation, while the ratio of the maximum load to the design load for each city varies considerably. The close range of the total loads to Pittsburgh is due to the fact that the maximum temperatures are approximately the same for the four cities. The wide variation in the maximum to design ratio is due to (1) the diurnal march of temperatures, (2) the seasonal temperature range, and (3) the relative frequency of occurrence of high temperatures.

The maximum to design ratio also shows that for Pittsburgh a system 50 per cent larger than that based on design conditions would be required

TABLE 4—MAXIMUM LOAD

	NEW YORK		PITTSBURGH		KANSAS CITY		NEW ORLEANS	
Load	Tons	Per Cent	Tons	Per Cent	Tons	Per Cent	Tons	Per Cent
Trans.	5.6	18.3	5.2	17.3	8.1	26.1	4.2	12.7
Vent.	14.0	46.0	13.9	45.7	13.5	40.8	18.0	54.5
Int. Load	10.9	35.7	10.9	37.0	10.9	33.1	10.9	32.8
Total	30.5	100.0	30.0	100.0	32.5	100.0	33.1	100.0
Ratio to Pittsburgh	1.02		1.00		1.08		1.10	
Max./Des.	1.38		1.51		1.27		1.15	

to take care of any possible weather condition which might exist. As the design dry-bulb is exceeded only 2½ per cent of the time, and the wet-bulb exceeded only 5 per cent of the time, a 50 per cent increase in investment cost is a large premium to pay to take care of these exceeded conditions and unless the problem is very special, the economics does not warrant the additional expenditure.

CONCLUSIONS

1. With the data given in the previously referred to paper and the analyses of design relations as herein given, the design conditions as specified are adequate for practically all problems in air conditioning, since the exceeded time of 2½ per cent for the dry-bulb and 5 per cent for the wet-bulb and dew-point are of no serious consequence in comfort cooling and commercial air conditioning problems.

2. For the design of chemical dehumidification equipment and cooling coils the coincident dry-bulb should be taken from the combination of design wet-bulb and design dew-point.

3. A further study should be made of solar radiation in cooperation with other Society Research Technical Advisory Committees studying this problem before any design conditions can be properly set up.

DISCUSSION

W. L. FLEISHER: As I expressed the appreciation of the Society to Mr. Albright for his great work, I think it is only fair that the Society should know that Mr. Everetts contributed greatly of his time and thought to the compilation of the work that Mr. Albright did, and I think Mr. Albright would be the first person to express his appreciation for the assistance that Mr. Everetts gave him. I feel the whole Society should be deeply grateful to Mr. Everetts for his work in behalf of this Society.

G. L. WIGGS: I notice here that they refer to these temperatures, the dry-bulb, as not being exceeded more than 2½ per cent of the time, and the wet-bulb not more than 5 per cent. Mr. Everetts shows in his paper that the variation in wet-bulb is much greater and yet they have seen fit to work to a smaller variation in dry-bulb. It seems to me that the condition should be reversed, although it may be conceded that something which is only exceeded 5 per cent of the time is quite satisfactory, but I would like to know why they used the two percentages.

JOHN EVERETTS, JR.: I am not going to answer that question because that does not come under my part of the paper, Mr. Wiggs. But those values are taken from the other paper.[1] In other words, the other paper set up certain design conditions, and the purpose of the paper which I just gave was merely to apply those design conditions. There is a very definite reason for picking those 2½ and 5 per cents for dry-bulb and wet-bulb, and I am going to ask Mr. Albright, under whose charge that particular question comes, to answer that.

J. C. ALBRIGHT: The 5 per cent value of the wet-bulb temperature has been chosen because such a value has proved adequate in actual practice over a period of years, as mentioned in the previous paper.[2] The reason why a 5 per cent value is satisfactory on the wet-bulb temperature is that such a value will be exceeded only a relatively short period in any one day. The fly-wheel action of a system will normally carry through a wet-bulb temperature condition in excess of the 5 per cent value without a serious over-run of temperatures.

Now, with regard to this 2½ per cent value on the dry-bulb temperature, all of the maximum dry-bulb temperatures take place during the 12 daylight hours, therein differing from the wet-bulb condition, since some of the maximum wet-bulb temperatures take place at night. The daily characteristic wet-bulb temperature curve has a very flat peak, whereas the dry-bulb characteristic curve has a very decided peak, coupled with a greater daily range. Consequently a smaller percentage of the time should be chosen for a dry-bulb percentage than for the wet-bulb percentage to introduce the correct protection against temperature over-runs.

From analysis of practice and other considerations, the value of 5 per cent of the daylight hours on dry-bulb would give comparable design conditions to a value of 5 per cent of the 24-hour data on the wet-bulb. Since all of the high dry-bulbs take place during the daytime, this is equivalent to saying that 2½ per cent of the 24 hours on the dry-bulb was giving an equal safety factor to 5 per cent of the 24 hours on the wet-bulb temperature.

With regard to dew-point, very little practical experience as to what to use is available. Less attention has been paid to the dew-point in the past, and to the

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[1] Analysis of Summer Weather Data in the United States, by J. C. Albright. (ASHVE TRANSACTIONS, Vol. 45, 1939, p. 397.)
[2] Loc. Cit. Note 1.

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close choice of it, than to the wet-bulb and dry-bulb temperatures. More attention should be paid to it, particularly with the growth of coil usage and chemical dehumidification. Such experience as exists indicates that the dew-point design temperature is no more critical than that of the wet-bulb temperature, hence the 5 per cent value is used for the dew-point temperature.

Summing up the shortage of the peaks, the fly-wheel action that will take place in a system, coupled with the fact that a maximum dry-bulb temperature probably will not occur simultaneously with a maximum of wet-bulb or dew-point temperature, makes the chosen values safe.

No. 1126

THE FLOW OF HEAT THROUGH WALLS

By F. E. GIESECKE,* (*MEMBER*), COLLEGE STATION, TEX.

THE size of the equipment necessary to air condition a building properly depends very largely on the construction of the building. For example, two buildings may be identical in other respects, but one may have heavy exterior walls through which heat flows slowly and in which large quantities of heat can be stored, while the other may have light exterior walls through which heat flows readily and in which only small quantities of heat can be stored. The former building can be air conditioned with a smaller installation and at a lower annual cost than the latter, because the heavier walls equalize to a greater extent the daily, as well as the seasonal variations in temperature and reduce thereby to a greater extent the maximum heating and cooling loads for which the air conditioning system must be designed.

In order to take proper account of the influence of the heat conductivity and of the heat storage capacity of the exterior walls, when heat-loss and heat-gain computations are made for a building, it is necessary to know how heat flows into, in, and out of such walls.

Ordinary heat-loss and heat-gain calculations for buildings are based on the assumption that the temperature gradient through a wall may be represented by a straight line. For example, Fig. 1, line *a*, represents the usual assumption of the heat flow through a 12-in. brick wall when the inside temperature is 70 and the outside temperature is 50 F. This assumption is only correct when the inside and outside temperatures remain constant and when they have been constant for a sufficient time for the flow of heat through the wall to have attained a steady state. Whenever the flow of heat through the wall is variable, the temperature gradient can only be represented correctly by a curved line, and, since this flow of heat can vary in an infinite number of ways, the temperature gradient may be represented by an infinite number of differently curved lines, as suggested by lines *b*, *c*, and *d*.

A thorough study of variable heat flow was first made by Prof. J. P. J. Fourier, and a fundamental mathematical treatment was developed by him and

* Director, Engineering Experiment Station, Texas A. & M. College.

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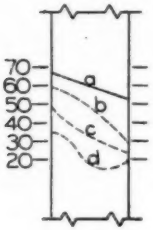

FIG. 1. THERMAL GRADIENTS

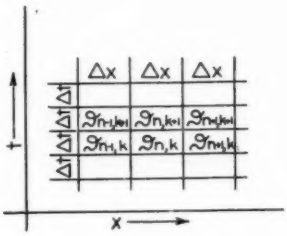

FIG. 2. THERMAL FIELD SHOWING VARIATION OF TEMPERATURE WITH TIME AND WITH DISTANCE

published in *Theorie Analytique de la Chaleur*, 1822. This exact mathematical treatment is very complicated and difficult to apply. For ordinary heating problems, it is sufficiently accurate to divide the time into definite intervals and to divide the wall into a definite number of slabs of equal thickness, and to assume that the flow of heat into and through the wall, and the temperatures within the wall change in a series of successive steps and remain constant during the respective intervals of time and space between the successive changes.

This method of studying variable heat flow was developed by Dr. Ernst Schmidt and his analysis published in Dr. August Foppls *Festschrift*, 1924.

The method is briefly as follows:

The differential equation of the flow of heat, in one direction only, in a wall built of isotropic material, as developed by Professor Fourier, is:

$$\frac{\delta \vartheta}{\delta t} = \frac{k}{cw} \frac{\delta^2 \vartheta}{\delta x^2} \quad \ldots\ldots (1)$$

where,

ϑ = temperature, degrees Fahrenheit.
t = time, hours.
k = thermal conductivity of wall, Btu per hour per degree Fahrenheit per foot.
c = specific heat of wall.
w = weight of wall, pounds per cubic foot.
x = coordinate, perpendicular to the surface of the wall, feet.

If small but definite units or increments of time and space Δt and Δx are substituted for the differentials δt and δx, and if the temperature remains constant during each time interval, equation (1) may be written with a sufficient degree of accuracy:

$$\frac{\Delta \vartheta}{\Delta t} = \frac{k}{cw} \frac{\Delta^2 \vartheta}{(\Delta x)^2} \quad \ldots\ldots (2)$$

By transforming and indicating the partial character of the temperature differences,

$$\Delta_t \vartheta = \frac{k}{cw} \frac{\Delta t}{(\Delta x)^2} \Delta^2_x \vartheta \quad \ldots\ldots (3)$$

It will be shown that, in order to apply Equation (3) in heat flow calculations, it is advantageous to select values for Δt and Δx which will reduce the terms $\frac{k}{cw}\frac{\Delta t}{(\Delta x)^2}$ to one-half. This can always be done by assuming Δx and solving for Δt or assuming Δt and solving for Δx in the equation:

$$\Delta t = \frac{cw}{2k}(\Delta x)^2 \quad (4)$$

For example, for a brick wall having $k = 0.5$, $c = 0.2$, and $w = 125$, if Δx is 2 in., Δt will be 41⅔ min. For a concrete wall, having $k = 1.0$, $c = 0.2$, and $w = 144$, if Δx is 2 in., Δt will be 24 min, or, for a wall of glass or mineral wool having $k = 0.0225$, $c = 0.2$, and $w = 1.5$, if Δx is 2 in., Δt will be 11⅙ min.

Assuming that the proper selection of Δt and Δx will always be made,

$$\Delta_t \vartheta = \frac{1}{2}\Delta^2_x \vartheta \quad (5)$$

In order to understand the meaning of the terms in Equation (5), examine the charts of Figs. 2 and 3.

The chart of Fig. 2 shows the thermal field in a portion of the vertical section of a wall as the temperature varies with respect to distance and with respect to time. One of the sections is designated $\vartheta_{n,k}$, this represents the temperature in the unit section which is removed $n\Delta x$ from the t-axis and $k\Delta t$ from the x-axis. The designations of the adjacent sections are similar.

It is evident from Fig. 2 that,

$$\Delta_t \vartheta = \vartheta_{n,k+1} - \vartheta_{n,k} \quad (6)$$

The chart of Fig. 3 shows the same thermal field as Fig. 2, but only for one unit of time; it shows how the temperature varies with respect to distance.

It is evident from Fig. 3 that,

$$\Delta_x{}^2 \vartheta = \vartheta_{n-1,k} + \vartheta_{n+1,k} - 2\vartheta_{n,k} \quad (7)$$

Substituting from Equations (6) and (7) in Equation (5)

$$\vartheta_{n,k+1} = \frac{1}{2}(\vartheta_{n-1,k} + \vartheta_{n+1,k}) \quad (8)$$

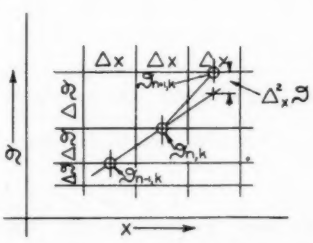

FIG. 3. THERMAL FIELD SHOWING VARIATION OF TEMPERATURE WITH DISTANCE ONLY

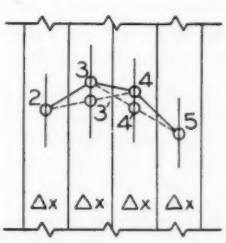

FIG. 4. METHOD OF PASSING FROM ONE THERMAL GRADIENT TO THE NEXT SUCCEEDING ONE

This shows that if the thermal gradient in a wall is known during any given unit of time k, the thermal gradient during the succeeding unit of time, $k+1$, is found by taking the arithmetic mean of the temperatures of alternate sections of the given thermal gradient and placing it equal to the temperature of the intermediate section in the required gradient. The required arithmetical mean may be found numerically or graphically. For example, if in Fig. 4 the line *2, 3, 4,* and *5* represents the thermal gradient in a wall during some one unit of time, the points *3'* and *4'*, of the thermal gradient during the next succeeding unit of time, may be found graphically as shown.

This method does not determine the initial point *2'* of the required thermal gradient. To find this point, it may be necessary to calculate (1) the quantity of heat flowing from the second section to the first section (knowing the

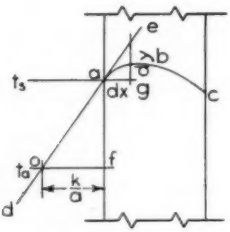

FIG. 5. METHOD OF LOCATING DIRECTING POINT OF TANGENTS TO THERMAL GRADIENTS

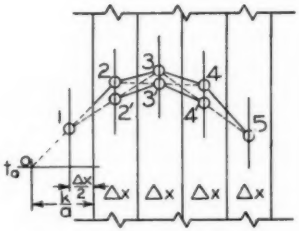

FIG. 6. METHOD OF PASSING FROM ONE THERMAL GRADIENT TO THE NEXT SUCCEEDING ONE

temperature difference and the thermal conductivity); (2) the quantity of heat flowing from the first section to the air (knowing the temperature difference and the film coefficient); (3) the net loss of heat by the first section; and (4) the reduction in temperature of the first section resulting from this net loss. However, instead of making such a long and complicated calculation, the required point can be located graphically with sufficient accuracy by a method based on the following reasoning:

In Fig. 5, let the curve *a b c* represent the temperature gradient in the wall, let *d e* be the tangent to this curve at the point *a;* let dy/dx be the slope of the tangent and therefore the slope of the temperature gradient at the surface of the wall.

Let t_s and t_a denote, respectively, the temperatures of the surface of the wall and of the air adjacent thereto, and let these temperatures be represented graphically as shown in Fig. 5. Let k denote the thermal conductivity of the material of the wall and a the film coefficient. Then the heat flowing from the surface of the wall to the surrounding air will equal $a\ (t_s - t_a)$ Btu per square foot per hour; this quantity of heat is equal to that which flows to the

outside wall surface from the wall section immediately adjacent thereto and which is $k \frac{dy}{dx}$. Hence, $a\ (t_s - t_a) = k \frac{dy}{dx}$ and hence, $\frac{dx}{dy}(t_s - t_a) = \frac{k}{a}$.

From the similar triangles $o\ f\ a$ and $a\ g\ e$, it is then evident that $o\ f = \frac{k}{a}$.

If the point o is called the directing point of the tangent to the temperature gradient at the surface of the wall, then: *The directing point is located on the*

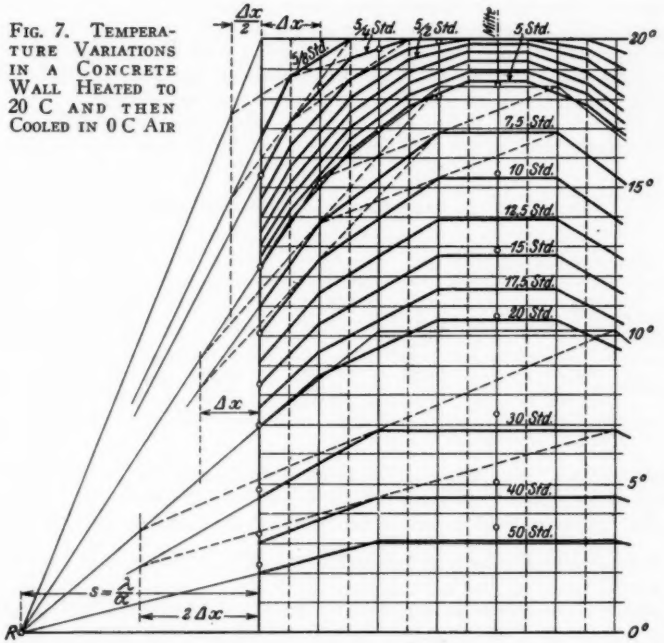

FIG. 7. TEMPERATURE VARIATIONS IN A CONCRETE WALL HEATED TO 20 C AND THEN COOLED IN 0 C AIR

line which represents the average air temperature, during the interval of time under consideration, and at a distance $\frac{k}{a}$ *from the surface of the wall.*

Applying this construction to Fig. 4, the point *2'* may be found as shown in Fig. 6, as follows: Draw the line t_a, representing the average temperature of air in contact with the wall; locate the directing point o on that line at a distance $\frac{k}{a}$ from the wall; join the points o and *2;* draw a line parallel to the surface of the wall at a distance $\frac{\Delta x}{2}$ from the wall; the point of intersection of this line with the line $o\ 2$ is the point *1* of the temperature gradient, ex-

tended beyond the surface of the wall; join points *1* and *3* to find point *2'*, as shown.

The points *6* and *5'* may be found in a similar manner.

The question will naturally arise, how accurate is this graphical method of determining the flow of heat in solids. To determine this, Dr. Schmidt assumed a concrete wall 15.75 in. thick, heated uniformly to 20 C, and then cooled in air at 0 C, and determined the rate of cooling. The results of this study are shown in Fig. 7, the calculated values by small circles and those secured graphically by straight lines. Both sets of values are given in Table 1.

It appears from this tabulation that, at the end of 20 hours, the difference between the two results varied only from about 1 to 1½ per cent. Since these results were secured with time increments varying from 37½ min to 2½ hours, it is evident that the graphical method is sufficiently accurate for calculations on which the design of air conditioning equipment is to be based.

The diagram of Fig. 6 may also be used to determine the quantity of heat flowing into the wall, or out of the wall, or from one portion of the wall to the

TABLE 1—WALL TEMPERATURES IN DEGREES CENTIGRADE

TIME IN HOURS		2½	5	10	20	50
Wall center	Fourier.....	19.70	18.46	15.48	10.68	3.52
	Schmidt.....	19.85	18.42	15.33	10.52	3.03
Wall surface	Fourier.....	13.98	12.30	10.08	6.96	2.30
	Schmidt.....	13.98	12.25	10.06	6.90	2.02

adjacent portions. In every case, heat flows from a region at higher to one of lower temperature. Since the point *2* represents a higher temperature than the point *1*, heat is flowing out of the wall into the adjacent air; the rate of flow is $k \frac{dy}{dx}$ Btu per hour per square foot. The value of k is known and the value of $\frac{dy}{dx}$ is shown by the slope of the line *2 1* with the horizontal and is expressed in degrees per foot or degrees per inch, depending on the scales used for the drawing. If point *2* were lower than point *1*, heat would be flowing from the adjacent air into the wall.

To explain the method of determining the quantity of heat flowing in a wall, consider Fig. 8 and let the lines *A B* and *C D* represent the temperature gradients at the beginning and end of a definite heating period—for example, nine hours. The area *A B C D* will then represent the heat lost by the wall during the nine-hour period. If the average difference of the temperatures shown by the two temperature gradients *A B* and *C D* is 1.4 deg, the wall will have cooled 1.4 deg. If the wall is 12 in. thick, weighs 125 lb per cubic foot, and has a specific heat of 0.2, the heat lost by the wall will be $125 \times 0.2 \times 1.4$, or 35 Btu in the nine-hour period.

If the line *E F* represents the average of the tangents drawn to the temperature gradients at the inner face of the wall, and if its slope is 0.8 deg

per inch, and if the thermal conductivity of the brick wall is 6 Btu per hour per degree Fahrenheit per inch, the quantity of heat which will have entered the wall at $A\ C$ is $k\ \frac{dy}{dx}$ or $6 \times 0.8 \times 9$, or 43.2 Btu in the nine-hour period.

If the line $G\ H$ represents the average of the tangents drawn to the temperature gradients at the outer face of the wall, and if its slope is 1.5 deg per inch, the quantity of heat which will have passed out of the wall at $B\ D$ is $6 \times 1.5 \times 9$, or 81 Btu in the nine-hour period.

Evidently, the quantity of heat passing out at $D\ B$ (81) must be equal to the sum of the quantities entering the wall at $A\ C$ (43.2) and that lost by the wall during the same period (35). In the case shown in Fig. 8, the quantity leaving the wall is 81 and the sum of the two other quantities is 78.2, or less by 2.8 Btu. This shows that there is an error in the assumptions made in the

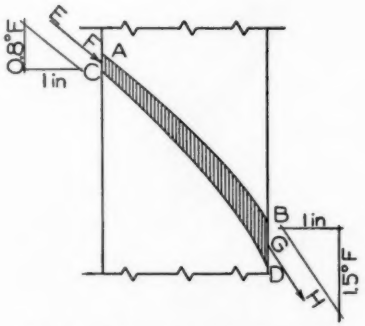

FIG. 8. METHOD OF DETERMINING HEAT LOST OR GAINED BY A WALL

construction of Fig. 8. One or more of the three calculated values, 43.2, 35, and 81, must be incorrect. If it is assumed that the correct value of the loss of heat by the wall is the average of the two calculated values, or 79.6, the error will be about 1.8 per cent.

To show the practical application of the method described, determine the flow of heat during a 24-hour period through the 12-in. brick wall shown in Fig. 9. The overall heat transmission coefficient of this wall is:

$$U = \frac{1}{\frac{1}{1.65} + \frac{12}{6} + \frac{1}{6}} = 0.361$$

Assume that the indoor air temperature remains at 70 F and the outdoor temperature begins with 50 F at 6 p.m. and varies according to the chart of Fig. 9B.

Assume that the indoor and outdoor temperatures of 70 and 50 F have remained constant for a sufficient time so that the initial temperature gradient can be represented by a straight line.

Draw (see Figs. 9A and 9B) (1) a section of the wall *A B C D;* (2) the

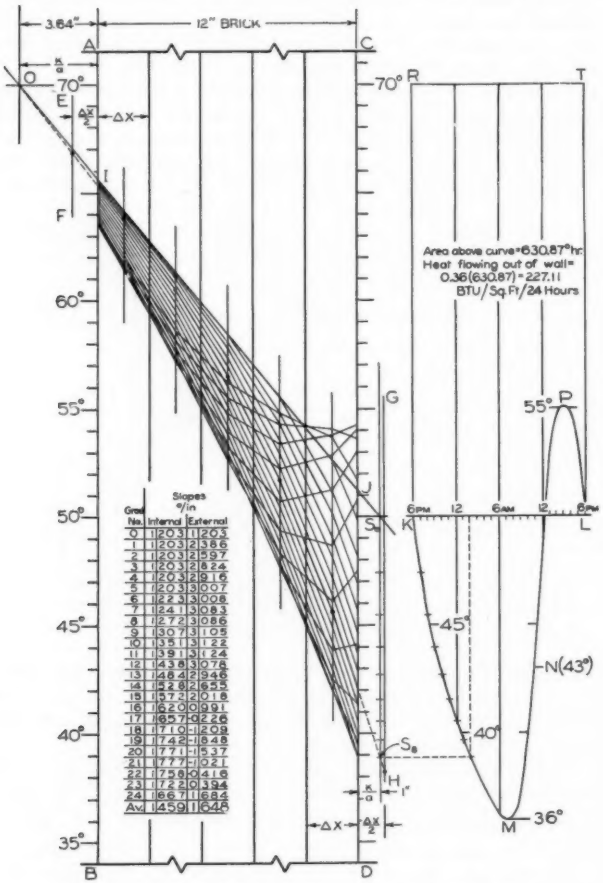

FIG. 9A FIG. 9B

70 and 50 F temperature lines *R T* and *K L;* (3) the temperature chart *K M N P L* for the 24-hour period; (4) vertical lines dividing the wall into five slabs of equal thickness (2.4 in.); (5) the median line of these slabs; (6) the two lines *E F* and *G H* parallel to the faces of the wall and at a distance $\frac{\Delta x}{2}$ from them, and (7) the initial temperature gradient IJ.

To draw the initial temperature gradient, proceed in the usual way; the three consecutive resistances to heat flow through the wall are $\frac{1}{1.65}$, $\frac{12}{6}$, and $\frac{1}{6}$; the three consecutive temperature drops are proportional thereto, their

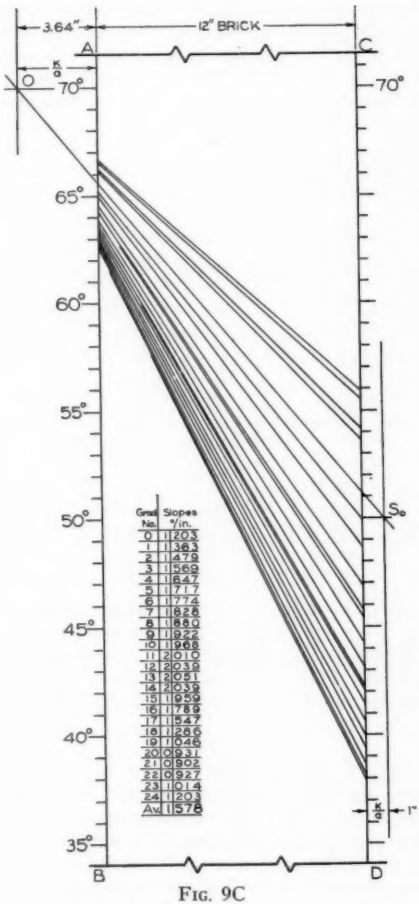

FIG. 9C

(Left to Right)

FIG. 9A. ACCURATE DETERMINATION OF THE FLOW OF HEAT THROUGH A 12-IN. BRICK WALL DURING A 24-HOUR PERIOD

FIG. 9B. TEMPERATURE AND DEGREE-HOUR CHART

FIG. 9C. APPROXIMATE DETERMINATION OF THE FLOW OF HEAT THROUGH A 12-IN. BRICK WALL DURING A 24-HOUR PERIOD

sum is 20 deg, and they are, therefore, 4.4, 14.4, and 1.2 deg. Set off, in the outer faces of the wall, the two points $70 - 4.4$ or 65.6 and $50 + 1.2$ or 51.2, and draw the temperature gradient IJ through these points.

Or, better and quicker, calculate the distances of the two directing points O and S_0 from the faces of the wall, $\frac{0.5 \times 12}{1.65}$ or 3.64 in., and $\frac{0.5 \times 12}{6}$ or

1.00 in., set them off as shown and draw the initial temperature gradient through them.

Having assumed Δx as 2.4 in., Δt will be found by Equation (4) to be 60 min. Set off 60-min points along the line K L of the temperature chart, draw vertical lines through the division points to intersect the temperature chart and, from the points of intersection, draw horizontal lines to intersect the vertical line through the directing point S_0, thus determining the consecutive positions of the directing point S_1, S_2, etc. Since the indoor temperature is to remain 70 F, the directing point O does not move from its original position.

Having drawn the initial temperature gradient and having the permanent location of the directing point O, and the consecutive locations of the directing

TABLE 2—HEAT LOSSES THROUGH WALLS

DIVISIONS	FIVE	SIX	SEVEN
Heat Received from Inside	210.1	211.3	212.8
Heat Lost by Wall	35.0	34.3	34.9
Sum of the Two	245.1	245.6	247.7
Heat Lost to Outside	237.3	235.2	235.5
Differences	7.8	10.4	12.2

point S, draw the required 24 temperature gradients as shown and as explained for Figs. 4 and 5. For example, at 2 a.m. the outdoor temperature is about 39 F, the directing point is S_8, and the temperature gradient is the one marked by small circles. As each successive temperature gradient is drawn, measure the slopes of the two tangents and record them as shown in the table in Fig. 9A.

Find the averages of these slopes (by Simpson's Rule) to be, respectively, 1.459 and 1.648 deg per inch. Accordingly, the heat entering the wall from the room is 210.1 Btu and the heat leaving the wall to the outdoor air is 237.3 Btu in the 24-hour period. Next, and as explained for Fig. 8, determine the average drop in temperature of the wall, as shown by the difference between the initial and final temperature gradients, to be 1.40 deg and, from that, the loss of heat by the wall during the 24-hour period to be 35 Btu.

Adding the 35 Btu lost by the wall to the 210.1 Btu received by the wall from the room, gives 245.1 Btu as the total loss of heat by the wall. However, according to the tangents to the several temperature gradients at the outer face of the wall, the total loss is 237.3 Btu. The difference of 7.8 Btu between these two values represents an error in the construction. If the true value

of the heat loss is the average of the two determined values, the error of construction is 1.6 per cent, which is sufficiently low to permit the use of this graphical method in engineering calculations.

However, it seems that if the wall were divided into slabs thinner than 2.4 in. and the time divided into units shorter than 60 min, the error should be smaller, and that the error should approach zero as the space and time intervals are made smaller and smaller.

To try this out, diagrams were constructed like that of Fig. 9 but with the wall divided into six and seven slabs, respectively, so that the slab thicknesses were 2 and 1⅚ in. and the time intervals were 41⅔ and 30$^{30}/_{49}$ min, respectively. Calculations were made as for Fig. 9, and the results, corrected to the 24-hour period, were secured as given in Table 2.

The results of these three determinations are so nearly identical that the differences may be accounted for by inaccuracies in the graphical constructions. It follows that there may be a considerable variation in the thickness of the slabs into which the wall is divided without materially affecting the results. This conclusion is supported by the data of Fig. 8 in which the slab thickness was doubled without producing serious inaccuracies.

To test the accuracy of the graphical construction from another angle, connect the outdoor temperature chart *K M N P L* of Fig. 9B to the indoor 70 F temperature line *R T*.

The chart *K R T L P N M* shows the number of degree hours, 630.87, during the 24-hour period. The heat loss through the wall, as usually calculated, is 630.87×0.36 or 227.1 Btu in 24 hours, per square foot of wall surface. This is 14.1 Btu or about 6 per cent less than that determined graphically. The question arises, which of the two values is more nearly correct.

To study this question, construct the 25 thermal gradients, corresponding to the 25 thermal gradients of Fig. 9A, as shown in Fig. 9C, in accordance with the formula:

$$H = A(t - t_o)$$

where,

H = Heat transmitted, Btu.
A = Area of wall, square feet.
= Over-all coefficient of heat transmission, Btu per hour per square foot per degree Fahrenheit.
t = Temperature of inside air, degrees Fahrenheit.
t_o = Temperature of outside air, degrees Fahrenheit.

These gradients indicate that the quantity of heat flowing out of the wall at any instant is equal to that flowing into the wall at that instant.

By measuring the slope of each gradient and recording it, as shown in Fig. 9C, calculating their average 1.578 and multiplying by K the conductivity,

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6, and by t the time, 24 hours, it is found that the heat flowing through the wall is 227.2 Btu in 24 hours, or the same as that determined from the degree-hour chart of Fig. 9A, as it should be.

Evidently, the calculated value of 227.2 Btu is based on the assumption that during every instant the quantity of heat flowing out of the wall is equal to

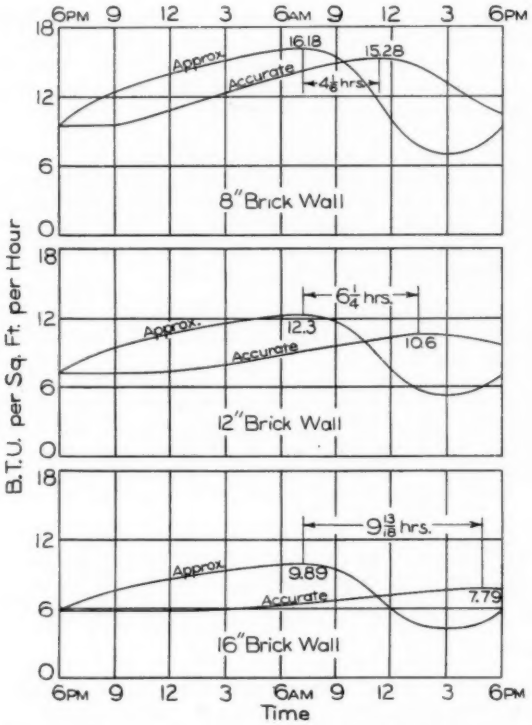

FIG. 10. VARIATION IN TIME-LAG OF HEAT FLOW THROUGH 8-IN., 12-IN., AND 16-IN. BRICK WALLS

the quantity of heat flowing into it. This would be approximately true in a wall consisting only of a thin sheet of metal having practically no heat storage capacity. In all walls, as ordinarily built, there is a considerable heat storage capacity, and the heat stored in the wall varies continuously as the indoor or outdoor air temperatures vary and as the temperature of the wall is influenced by solar radiation. For this reason, the quantity of heat flowing out of the wall will generally vary materially from that flowing in at the same instant.

For the particular conditions assumed for the 24-hour period under study, the average temperature of the wall was higher at the beginning than at the end of the 24-hour period. The quantity of heat lost by the wall in cooling during the 24-hour period, about 35.2 Btu, must be included in the heat loss calculations for the building.

Having completed the graphical studies of the flow of heat through a 12-in. brick wall as shown in Figs. 9A and 9C, similar studies were made for an 8-in. and for a 16-in. brick wall. The final results of these studies are shown in Fig. 10. Hereafter, the graphical method (Fig. 9A) will be designated the accurate method, because it is based on the heat storage capacity as well as on the heat conductivity. The method in general use at this time (Fig. 9C) will

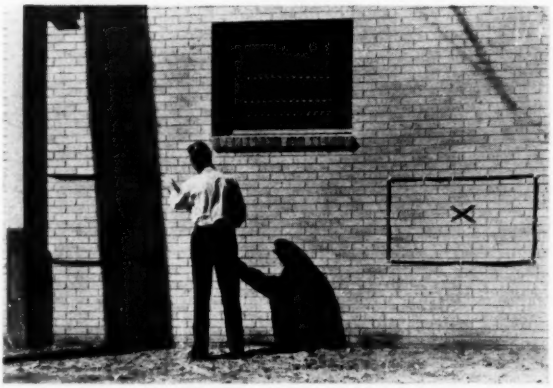

FIG. 11. WALL USED IN EXPERIMENTAL DETERMINATION OF HEAT FLOW

be designated the approximate method because it is based entirely on the heat conductivity of the materials composing the wall.

To compare the results secured by the accurate method with those secured by the approximate method, construct Fig. 10 to show the varying rates of heat flow from the room into the wall based on the slopes of the respective tangents as recorded in Figs. 9A and 9C for the 12-in. wall.

It appears from Fig. 10 that the maximum flow of heat from the room into the wall is less according to the accurate method than according to the approximate method, and that considerable time passes before the maximum flow occurs. This time lag increases with the thickness of the wall and varies from about 4 hours for an 8-in. wall to about 10 hours for a 16-in. wall.

Time lag has a very important bearing on heat flow calculations, especially when the cooling load resulting from solar radiation is to be determined.

In the past, heat flow calculations for buildings were restricted largely to the determination of heating loads and large portions of walls were replaced by

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windows and doors so that the heat loss through walls formed a comparatively small portion of the total heat loss of a building, and, consequently, the use of a scientifically incorrect method of heat flow calculation caused no serious harm.

At present, and probably for some time in the future, conditions are and will be quite different. With present-day improved lighting and ventilating systems, the proportion of wall spaces occupied by windows and doors is greatly reduced and the calculated heat loss through walls forms a much larger portion of the total heat loss than it did formerly. In addition, with the advent of

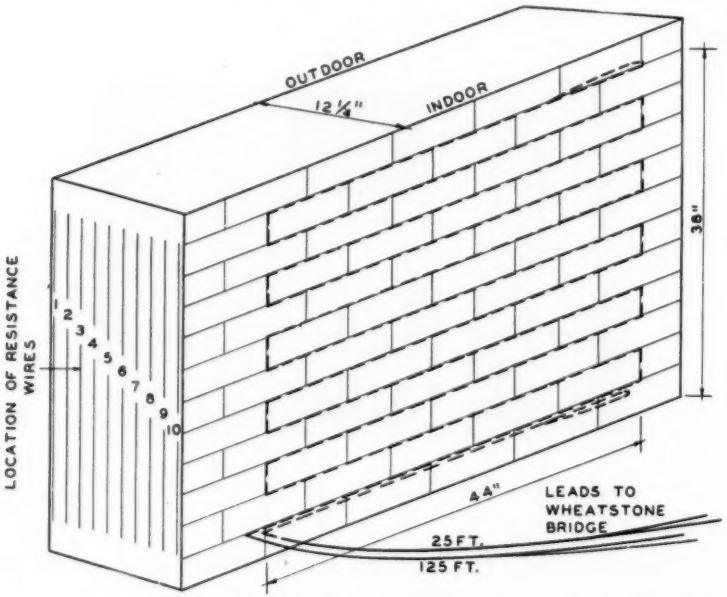

FIG. 12. ARRANGEMENT OF RESISTANCE WIRES IN WALL FOR EXPERIMENTAL DETERMINATION OF HEAT FLOW

air conditioning, heat flow calculations to determine cooling loads are probably more common and more important than those to determine heating loads, and it is more important than ever before to know the time lag in the flow of heat through walls which results directly from the heat storage capacities of the walls. For these reasons, it is important to abandon the present method of heat flow calculation and to develop a method which takes into account the thermal capacities as well as the thermal conductivities of the materials composing the walls.

When this shall have been done, a better opportunity than now exists will be provided for the cooperation of the architect and the heating engineer, so as

to produce buildings with those types of walls, floors, and roofs which, when combined with the necessary air conditioning equipment, will give the owner the most economical plant in first cost as well as in cost of operation and maintenance.

In order to make the calculations presented in this paper as simple as possible, they were based on a wall of uniform material throughout. In actual building construction, walls are of a composite nature.

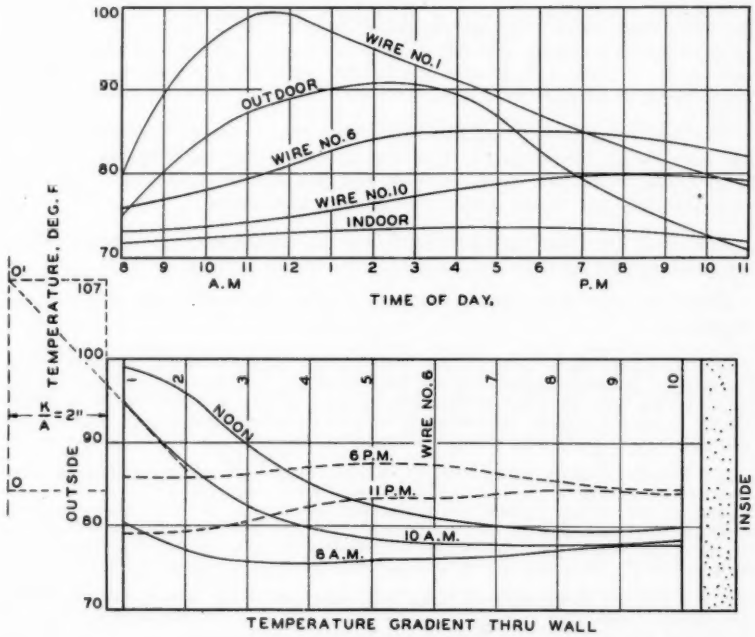

FIG. 13. TEMPERATURE CURVES AND TEMPERATURE GRADIENTS DERIVED IN EXPERIMENTAL DETERMINATION OF HEAT FLOW THROUGH WALL

The flow of heat through such composite walls can be traced by the graphical method presented above, but the construction is slightly more complicated. Illustrative examples are not submitted for lack of space and time.

So far, the discussion has been based entirely on theoretical reasoning. To compare the results secured with those which may be expected in actual installations, resistance wires were placed in a 13-in. brick wall so that the flow of heat into and through the wall could be measured. The wall is shown in Fig. 11; the resistance wires were placed in the two wall sections marked by

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black ribbons. Two sets of wires were placed so that by shielding one section of the wall from the sun's rays while the other section was exposed thereto, simultaneous temperature readings could be taken in the two sections and, in that way, the effect of direct solar radiation could be determined.

The wall was the southwest wall of an addition to the A. & M. College Exchange Store, erected in August 1938, and was built of common gray brick laid in cement-lime mortar.

Each set of wires (Fig. 12) consisted of 10 pairs of No. 20 enamel-covered copper wire; the two outer pairs were within ½-in. of the wall surface and the other eight pairs were spaced equally between them. Each pair consisted of one wire 125 ft long and one 25 ft long so as to eliminate end conditions and so as to have 100 ft of wire in the plane of the wall in which the temperature was to be measured.

The first temperature readings were taken about six weeks after the wall had been built. The upper chart of Fig. 13 shows how the temperature varied from 8 a.m. to 11 p.m. in Wires 1, 6, and 10; and also how the temperature varied during that period within the building and outdoors.

The lower chart of Fig. 13 shows five of the temperature gradients through the wall as determined by means of the 10 resistance wires.

Much valuable information can be secured from the two charts. For example:

(1) The upper chart shows clearly how the heat wave enters the outer surface of the wall, is gradually reduced by the heat absorption of the wall, and reaches the inner surface very much diminished, since Wire 1 attains its maximum temperature, 99, at 11:30 a.m., Wire 6 attains its maximum temperature, 85, at 6 p.m., and Wire 10 attains its maximum temperature, 80, at 9 p.m.

(2) The lower chart shows that, at 10 a.m., heat is flowing into the wall from outdoors at a rate of 36 Btu per hour per square foot (since the slope of the tangent to the 10 a.m. gradient is 6 deg per inch which, for a conductivity of 6, gives 36 Btu) and flowing out of the wall into the building at a rate of 10 Btu per hour per square foot (since the wall surface temperature is 78 and the room temperature 72; the difference, 6, with a film coefficient of 1.65, gives 10 Btu).

(3) When the temperature readings were taken the wind velocity was very low, not above 5 mph; for this velocity, the outdoor film coefficient is about 3. For this value and a value of 6 for the conductivity of the brick work (Fig. 5) the directing point of the gradient tangent is 2 in. from the face of the wall, and at 10 a.m. it is on the 84 F temperature line.

It appears from the lower chart that the 10 a.m. gradient tangent extended would intersect the directing point line near the 107 F temperature line, or about 23 deg above the usual location of this directing point. This shows that, for this particular wall and for this particular time, it is necessary to add approximately 23 deg to the outdoor temperature in order to account for the additional heat which enters the wall as a direct result of thermal radiation, if the heat flow calculation is to be made in the usual manner.

In studying these charts it must be borne in mind that the brick wall was only six weeks old when the temperature readings were taken. It is very likely that chemical and physical changes resulting from the setting of the mortar

were still taking place and were probably affecting the temperature distribution within the wall. For this reason, observations were discontinued and will be resumed during the summer of 1939 when it should be possible to secure valuable data from this set-up.

SUMMARY

1. Attention is called to the importance of considering the heat storage capacities as well as the heat conductivities in calculations relating to the heating and cooling loads for air conditioning.
2. An accurate graphical method, developed by Dr. E. Schmidt of Danzig, for tracing the flow of heat in and through walls is presented.
3. The flow of heat through a 12-in. brick wall during a 24-hour period is determined by the graphical method.
4. The variation in time-lag in the flow of heat with different types of walls is shown.
5. Preliminary results of an experimental study of the flow of heat through a 12-in. brick wall are given.

ACKNOWLEDGMENTS

The author is indebted to R. O. Loving, Instructor in Engineering Drawing, Texas A. & M. College, W. H. Badgett, Research Assistant, Engineering Experiment Station, Texas A. & M. College, and G. C. Kershaw, Laboratory Assistant, Engineering Experiment Station, Texas A. & M. College, for valuable assistance. Mr. Loving prepared all graphical constructions and made all calculations and drawings incident thereto. Mr. Badgett and Mr. Kershaw prepared the resistance wires and placed them in the wall as shown in Fig. 12; they also made the temperature observations and constructed the charts of Figs. 12 and 13.

DISCUSSION

J. E. RYAN:[1] Professor Giesecke has done an admirable piece of work and has introduced a novel method for this subject which both of us have addressed in our papers. The particular advantage of his paper is in its ability to handle nonhomogeneous wall construction which, although it is not brought out directly in the paper, has a sufficient outline given to enable anyone, with a little study, to carry it out.

The chief limitation that I see is that you must start at some time with a known temperature distribution. He started with a straight line, and, if I remember correctly, at about 6:00 a.m. Is not that correct?

PROF. F. E. GIESECKE: No, in the afternoon.

MR. RYAN: From the curves of temperature which he showed from actual measurement, I do not know if we could conclude that that is an altogether reasonable approximation. Truly, we must make one or another approximation, and the real check in the matter, when the wall is exposed to a periodic heat flow, is that at the end of his 24-hour cycle one should come back with the same temperature profile

[1] Engineering General Dept., General Electric Co., Schenectady, N. Y.

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through the wall as at the beginning. If you are a good guesser in the first place that is what you should get. I think his curve came back quite close, but there is always that limitation.

Professor Giesecke's observations with regard to solar radiation and the effect on his directing point temperature are very interesting, and I commend them to everyone to study so that you may see how important this solar radiation is, despite the fact that we do not know too much about values in various localities.

PROFESSOR GIESECKE: We could have modified the outdoor temperature curve so that the heat stored in the wall was the same at the end as at the beginning of the 24-hour period, but that is not necessary. The area included between the initial and final temperature gradients and the sides of the wall represents the change in the quantity of heat stored in the wall during that period. The graphical construction is correct if, for any definite period of time, the change in the quantity of heat stored in the wall is equal to the difference between the quantity of heat flowing into the wall from the room and the net quantity of heat flowing out of the wall into the outer space, during that period, as determined by the graphical construction.

This paper is the result of cooperative research with E. K. Campbell in the study of intermittent heating; I hope to show in a future paper that this graphical construction is of great value in the study of intermittent heating.

No. 1127

REACTIONS OF OFFICE WORKERS TO AIR CONDITIONING IN SOUTH TEXAS

By A. J. RUMMEL,* SAN ANTONIO, TEX., F. E. GIESECKE,** W. H. BADGETT,*** (*MEMBERS*), AND A. T. MOSES,† COLLEGE STATION, TEX.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the Texas Engineering Experiment Station of the Agricultural and Mechanical College of Texas and the San Antonio Public Service Co., conducted under the supervision of the Research Technical Advisory Committee on Sensations of Comfort.

IN THE past there has been some disagreement among air conditioning engineers as to the most desirable temperature and humidity conditions to be maintained indoors in relation to various outdoor conditions. As a result of the research work done by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS at its Research Laboratory in Pittsburgh and in cooperation with the Ontario Research Foundation of Toronto and the Texas Engineering Experiment Station at College Station,[1, 2, 3, 4, 5, 6, 7] these differences of opinion have been practically eliminated for most localities. However, the optimum indoor conditions for warm climates where the average yearly temperature is fairly high have remained more or less a matter of conjecture

* Air Conditioning Engineer, San Antonio Public Service Co., San Antonio, Tex.
** Director, Texas Engineering Experiment Station, College Station, Tex.
*** Research Associate, Texas Engineering Experiment Station, College Station, Tex.
† Technical Assistant, Texas Engineering Experiment Station, College Station, Tex.

[1] Comfort Standards for Summer Air Conditioning, by F. C. Houghten and Carl Gutberlet. (ASHVE TRANSACTIONS, Vol. 42, 1936, p. 215.)

[2] Cooling Requirements for Summer Comfort Air Conditioning, by F. C. Houghten, F. E. Giesecke, C. Tasker, and Carl Gutberlet. (ASHVE TRANSACTIONS, Vol. 43, 1937, p. 145.)

[3] Summer Cooling Requirements of 275 Workers in an Air Conditioned Office, by A. B. Newton, F. C. Houghten, Carl Gutberlet, and R. W. Qualley. (ASHVE TRANSACTIONS, Vol. 44, 1938, p. 337.)

[4] Cooling Requirements for Summer Comfort Air Conditioning in Toronto, by C. Tasker. (ASHVE TRANSACTIONS, Vol. 44, 1938, p. 549.)

[5] Shock Experiences of 275 Workers After Entering and Leaving Cooled and Air Conditioned Offices, by A. B. Newton, F. C. Houghten, Carl Gutberlet, R. W. Qualley, and M. C. W. Tomlinson. (ASHVE TRANSACTIONS, Vol. 44, 1938, p. 571.)

[6] General Reactions of 274 Office Workers to Summer Cooling and Air Conditioning, by F. C. Houghten, A. B. Newton, R. W. Qualley, and Edward Witkowski. (ASHVE TRANSACTIONS, Vol. 44, 1938, p. 591.)

[7] Seasonal Variations in Effective Temperature Requirements, by F. E. Giesecke and W. H. Badgett. (ASHVE TRANSACTIONS, Vol. 44, 1938, p. 559.)

Presented at the 45th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Pittsburgh, Pa. January, 1939.

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and have been the subject of considerable discussion and even lawsuits. In order that this element of doubt might be removed, San Antonio, Texas, situated in one of the warmer regions of the country and with an average yearly temperature of 69.7 F, was chosen by the Technical Advisory Committee on Sensations of Comfort as a desirable location for a study of comfort conditions for office workers during the summer of 1938.

Outdoor temperatures in San Antonio as recorded by the United States Weather Bureau during July and August, 1938, the period in which this study was made, are given in Table 1.

The normal average 24-hour mean dry-bulb temperature from April 1 to October 1 is 78.8 F; from October 1 to April 1, it is 60.6 F, with a year around average of 69.7 F.

TABLE 1—WEATHER BUREAU DATA, SAN ANTONIO, TEXAS

TIME OF DAY		TEMPERATURES, DEGREES FAHRENHEIT					
		July		August		Average for July and August	
		D.B.	W.B.	D.B.	W.B.	D.B.	W.B.
8:00 A.M.	Min.	74.0	73.0	74.0	70.0		
	Max.	81.0	75.0	82.0	75.5		
	Avg.	78.0	74.5	78.3	74.0	78.2	74.2
12:00 Noon	Min.	86.0	72.0	77.0	72.5		
	Max.	93.0	77.0	93.0	76.0		
	Avg.	88.8	74.5	89.1	74.0	88.9	74.3
5:00 P.M.	Min.	81.0	74.0	75.0	72.5		
	Max.	98.0	77.0	97.0	76.5		
	Avg.	93.2	75.0	91.9	74.5	92.6	74.7
Avg. 24 Hours	Mean	85.0	73.2	84.6	73.8	84.8	73.5

During the period of this study, the minimum dry-bulb temperature recorded between 8:00 a.m. and 5:00 p.m. was 74 F and the maximum was 100 F. During those hours, the dry-bulb temperature was 90 F or above approximately 48 per cent of the time and 95 F or above approximately 18 per cent of the time.

The highest dry-bulb temperature on record for the month of July in any year since 1885 is 106 F, and for the month of August it is 107 F.

The average daily percentage of possible sunshine during July and August, 1938, was 85 per cent.

Between the hours of 8:00 a.m. and 5:00 p.m. the minimum wet-bulb temperature recorded was 70 F, the maximum was 77 F, and the average 74.5 F. During these hours the wet-bulb temperature was 75 F or above approximately 38 per cent of the time.

The building selected for this study was the air conditioned offices of the San Antonio Public Service Co. shown in Fig. 1. It was chosen because of the large number of office workers employed under one management in an air

conditioned space; because these employees provided a typical cross-section of office workers in this locality; because of the possibility of accurate control of temperature and humidity conditions in the building; and because of the willingness of the company to cooperate in the work.

The personnel of the offices in which the study was made was limited to that on the second and third floors, shown diagrammatically in Fig. 2, and consisted of 94 persons, of whom 72 (77 per cent) were men and 22 (23 per cent) were women. The group varied in age from under 20 to past 60.

FIG. 1. AIR CONDITIONED OFFICE BUILDING OF THE SAN ANTONIO PUBLIC SERVICE CO. IN WHICH THE STUDY WAS MADE

During the summer the men did not wear their coats while in the office, and the women wore light summer dresses. Working hours were from 8:00 a.m. to 5:30 p.m., with a lunch period of an hour and a half.

In the preliminary observations of dry- and wet-bulb temperatures which were made with an electric indicating psychrometer at each of the 94 desks occupied by the subjects used in the study, it was found possible to divide the entire space on the second and third floors of the building into 25 zones in each of which the effective temperature was very uniform.

During the first week of the study, temperature observations were made twice each day in each of the 25 zones. By the end of the week it became evident that the space could be divided into 7 zones as indicated in Fig. 2, in each of which there was practically no variation in effective temperature.

In the large offices, particularly along the south side of the building, it was expected that on the desks near the outside wall the effective temperature would be a little higher than on the desks nearer the center of the building, due to

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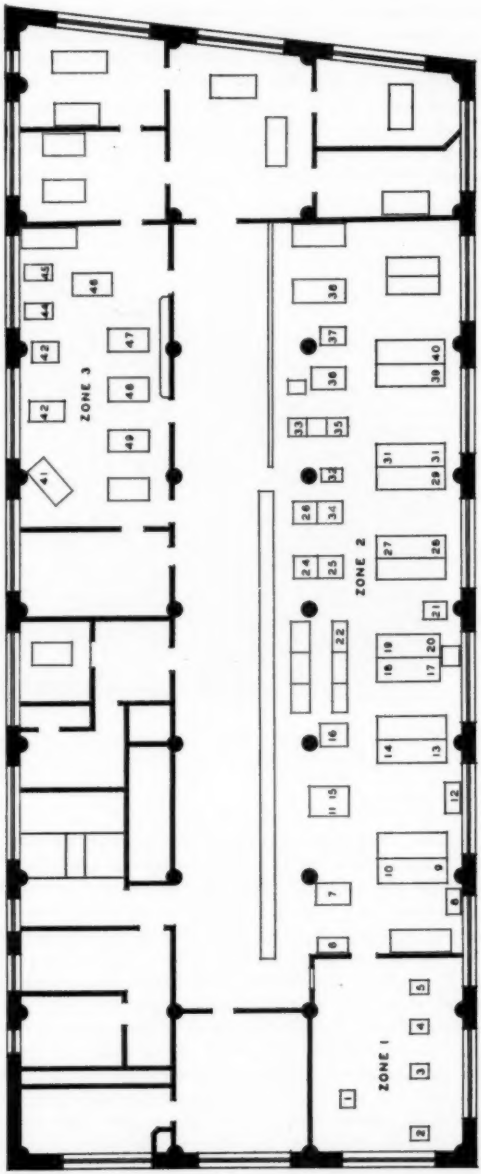


SECOND FLOOR
NOTE: NUMBERS IN RECTANGLES INDICATE LOCATION OF SUBJECTS

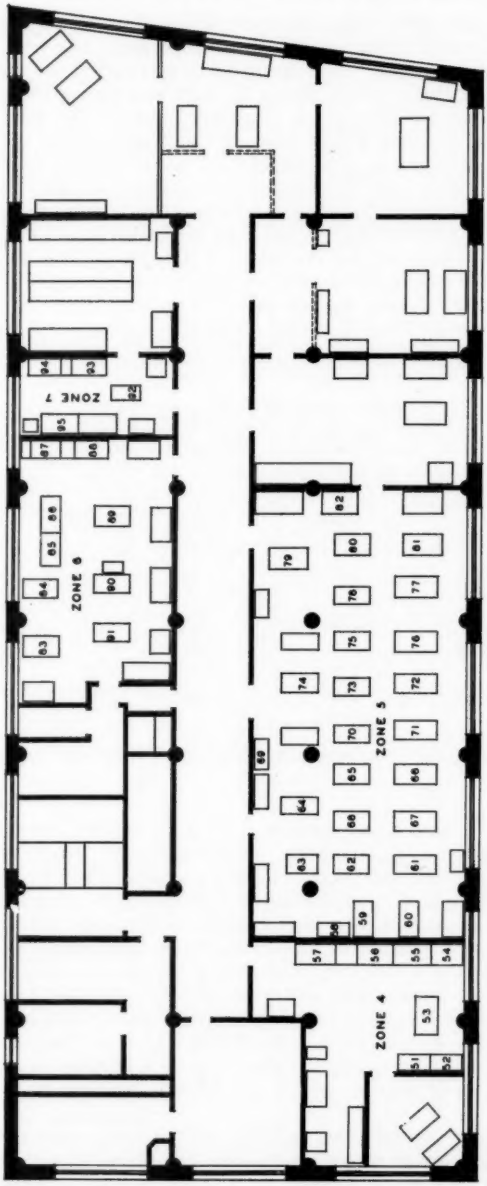
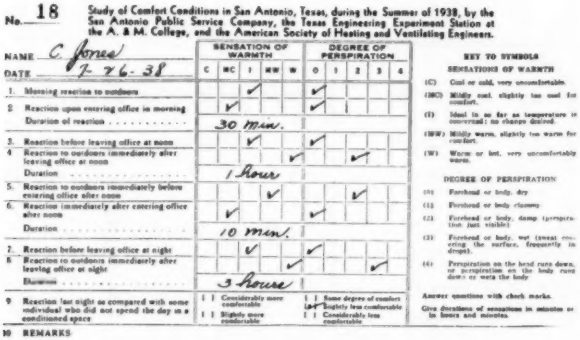


FIG. 2. AIR CONDITIONED OFFICE SPACE IN WHICH THE STUDY WAS MADE

radiation from the sun and to absorption of radiant heat by the wall itself. However, no significant difference in effective temperature was noted after a large number of observations. This unexpected uniformity of temperature distribution can be attributed, no doubt, to the fact that the air supply grilles are located quite high and the air is discharged toward the outside wall, striking the wall and dropping before reaching the breathing line in any part of the office. Venetian blinds are installed on all windows and their full length is used at all times. All illumination is supplied from indirect sources. The reactions of persons near the windows and walls exposed to the afternoon sun were compared with those in the center of the building and those on the shaded side of the building, and for the same effective temperature there seems to be very little difference, if any, in their sensations of warmth.

The actual collection of data for use in the study was begun on Monday, July 11, and was continued for 7 weeks, observations being made on 5 days



No. 18 Study of Comfort Conditions in San Antonio, Texas, during the Summer of 1938, by the San Antonio Public Service Company, the Texas Engineering Experiment Station at the A. & M. College, and the American Society of Heating and Ventilating Engineers.

NAME C. Jones
DATE 7-26-38

	SENSATION OF WARMTH					DEGREE OF PERSPIRATION				
	C	MC	I	MW	W	0	1	2	3	4
1. Morning reaction to outdoors			✓			✓				
2 Reaction upon entering office in morning		✓				✓				
Duration of reaction	30 min.									
3. Reaction before leaving office at noon			✓			✓				
4 Reaction to outdoors immediately after leaving office at noon				✓				✓		
Duration	1 Hour									
5. Reaction to outdoors immediately before entering office after noon				✓				✓		
6. Reaction immediately after entering office after noon		✓				✓				
Duration	10 min.									
7. Reaction before leaving office at night			✓			✓				
8 Reaction to outdoors immediately after leaving office at night					✓				✓	
Duration	9 Hours									

9 Reaction last night as compared with some individual who did not spend the day in a conditioned space
[ ] Considerably more comfortable
[ ] Slightly more comfortable
[ ] Same degree of comfort
[✓] Slightly less comfortable
[ ] Considerably less comfortable

10 REMARKS

KEY TO SYMBOLS

SENSATIONS OF WARMTH

(C) Cool or cold, very uncomfortable.
(MC) Mildly cool, slightly too cool for comfort.
(I) Ideal, in so far as temperature is concerned; no change desired.
(MW) Mildly warm, slightly too warm for comfort.
(W) Warm or hot, very uncomfortably warm.

DEGREE OF PERSPIRATION

(0) Forehead or body, dry
(1) Forehead or body clammy
(2) Forehead or body, damp (perspiration just visible)
(3) Forehead or body, wet (sweat covering the surface, frequently in drops).
(4) Perspiration on the head runs down, or perspiration on the body runs down or wets the body

Answer questions with check marks.
Give durations of sensations in minutes or in hours and minutes.

FIG. 3. DAILY QUESTIONNAIRE TO OFFICE PERSONNEL

each week. Each morning a questionnaire, as shown in Fig. 3, was given to each employee and at the same time the questionnaires for the preceding day were collected. In order that each individual might fully understand the purpose of this study, and in order that all interpretations of reactions might be as nearly uniform as possible, a certain amount of *coaching* was given each person during the first week. On the questionnaire each employee recorded his feeling of warmth at 8 different times during the day—immediately before and immediately after the morning and noon entrances into the building, and immediately before and after the noon and evening exits from the building. He also recorded his feeling of warmth during the preceding night. The information obtained from the questionnaires was used in studying the feeling of shock on entering or leaving an air conditioned building under various conditions.

Twice each day the dry-bulb and wet-bulb temperatures were observed in each zone and each occupant was asked to state his feeling of warmth at that moment, which was recorded on the data sheet shown in Fig. 4. This informa-

tion was used in the study of comfort conditions for office workers in an air conditioned building.

These data indicate rather definitely that, in spite of the mild winters and the intensity and duration of the summer heat in San Antonio, there is very little difference in the optimum effective temperature for indoor comfort in this locality as compared to other localities where similar investigations have been made. Because of the fact that the average outdoor temperature is much higher in the southern part of Texas than in the northern states, it has been commonly believed that indoor conditions somewhat warmer than the optimum for those localities would be entirely satisfactory. A previous study conducted

LOCATION 16

Study of Comfort Conditions in San Antonio, Texas, during the Summer of 1938, by the San Antonio Public Service Company, the Texas Engineering Experiment Station at the A. & M. College, and the American Society of Heating and Ventilating Engineers.

NAME Miss Hilma Eber PERIOD JUL 12 JUL 16

DATE	TIME	ROOM CONDITIONS				SENSATIONS OF WARMTH					DEGREE OF PERSPIRATION				
		DB	WB	ET	RH	C	HC	I	HW	W	0	1	2	3	4
12	10 00	82	69.0	76.0	52 %					✓		✓			
	3:18	82.7	68.9	76.0	50 %			✓			✓				
13	9:55	79.3	66.9	74.0	52 %			✓			✓				
	3 43	80.0	66.5	74.0	49 %			✓			✓				
14	9:56	77.3	65.2	72.0	52 %		✓				✓				
	3. 51	77.6	65.3	72.0	52 %		✓				✓				
15	9.50	80.0	68.0	74.5	53 %			✓				✓			
	3 54	80.5	67.5	74.5	50 %			✓			✓				
16	9 43	81.3	69.8	76.0	53%				✓			✓			
	3 45	78.3	66.0	73.0	52 %			✓			✓				

FIG. 4. DATA SHEET FOR RECORDING REACTIONS OF OFFICE PERSONNEL TO INDOOR ATMOSPHERIC CONDITIONS

at the Agricultural and Mechanical College of Texas at College Station, Texas [8] had indicated that such a belief was erroneous, but, because of the small number of subjects used in that study, and because of the fact that they were all young men and in good health, the results were not conclusive. The study just completed, however, has substantiated the results indicated by the Agricultural and Mechanical College of Texas study, and, for the sake of comparison, the San Antonio results have been plotted on a curve from the College Station study, as shown in Fig. 5.

There is no doubt that people who live in warm climates become acclimated, but the extent of that acclimatization is evidently not so great as it was formerly believed, and, even though inhabitants of warm climates can apparently withstand somewhat higher temperatures without suffering ill effects, they are not necessarily comfortable while doing so.

The study of the daily reactions of all employees indicates that optimum

[8] Loc. Cit. See Note 7.

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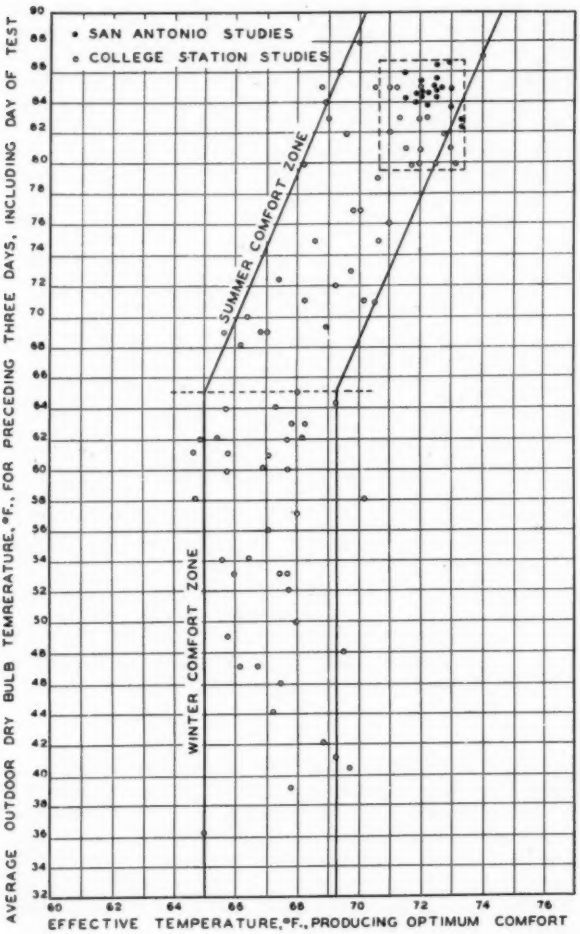

FIG. 5. COMPARISON OF THE SAN ANTONIO AND COLLEGE STATION, TEX., STUDIES. (EACH COLLEGE STATION POINT REPRESENTS THE REACTIONS OF 5 SUBJECTS AND EACH SAN ANTONIO POINT REPRESENTS THE REACTIONS OF 94 SUBJECTS)

indoor conditions are not appreciably affected by short-period fluctuations in the outdoor temperature.

The results of the study of comfort conditions are shown graphically in Fig. 6, in which the percentages of all persons who were comfortable after an hour or more in the air conditioned space are plotted against the effective temperatures at which the observations were made. These effective temperatures are based on air movements in the conditioned space not exceeding 20 fpm.

A superficial examination of the curve would indicate that an effective temperature of 71.5 deg would be ideal insofar as comfort is concerned. However, a more careful analysis of the results shows that at 71.5 deg ET, even though approximately 92 per cent of the subjects were comfortable, the majority of those not expressing a feeling of comfort were mildly cool, which is not desirable. At 72 deg ET approximately 90 per cent were found to be comfortable, 5 per cent mildly cool, and 5 per cent mildly warm. For this condition the relative humidity varied from a minimum of 44 per cent to a maximum of 59 per cent, remaining the greater part of the time between 48 and 53 per cent, with an average of 51 per cent.

A study of the reactions of all the women as compared to the reactions of all the men indicated that the optimum effective temperature for women is about 2 deg higher than the optimum for the men (Figs. 7, 8, and 9). This seems to apply equally to all age groups. The difference in summer comfort requirements for men and for women may be due partly to the difference in the amount of clothing worn.

In any application of air conditioning for human comfort its effect on health must be considered, and the fact cannot be denied that if a person is afflicted with any slight respiratory ailment the condition is aggravated by a feeling of chilliness. Therefore, for any large group of persons of both sexes an optimum effective temperature of about 73 deg is indicated. At 73 deg ET approximately 85 per cent of all persons were comfortable after having been in the building for one hour or more, and of those who were not comfortable only a small percentage was slightly too cool.

In the studies which were made at the Agricultural and Mechanical College of Texas [9] continuously over a period of a year, it was found, as shown in Fig. 5, that the optimum condition of maximum comfort for outdoor temperatures averaging over 80 F varied from about 70.5 deg ET to 73 deg ET with a mean of about 71.5 deg ET. In view of the fact that the subjects in the College Station study wore light-weight coats, whereas those in the San Antonio study did not, there is a remarkable agreement between the two studies when it is considered that wearing a coat makes one comfortable at from 1 to 2 deg lower effective temperature.

The reactions of the different age groups indicate that for both men and women past 40 years the optimum effective temperature is about one-half a degree higher than for those under 40 years as indicated in Figs. 10 and 11.

An attempt was made to determine the effect of health and stature on comfort. The data obtained were not adequate for any definite conclusions, but a few facts were indicated. For all persons whose health would not be

[9] Loc. Cit. See Note 7.

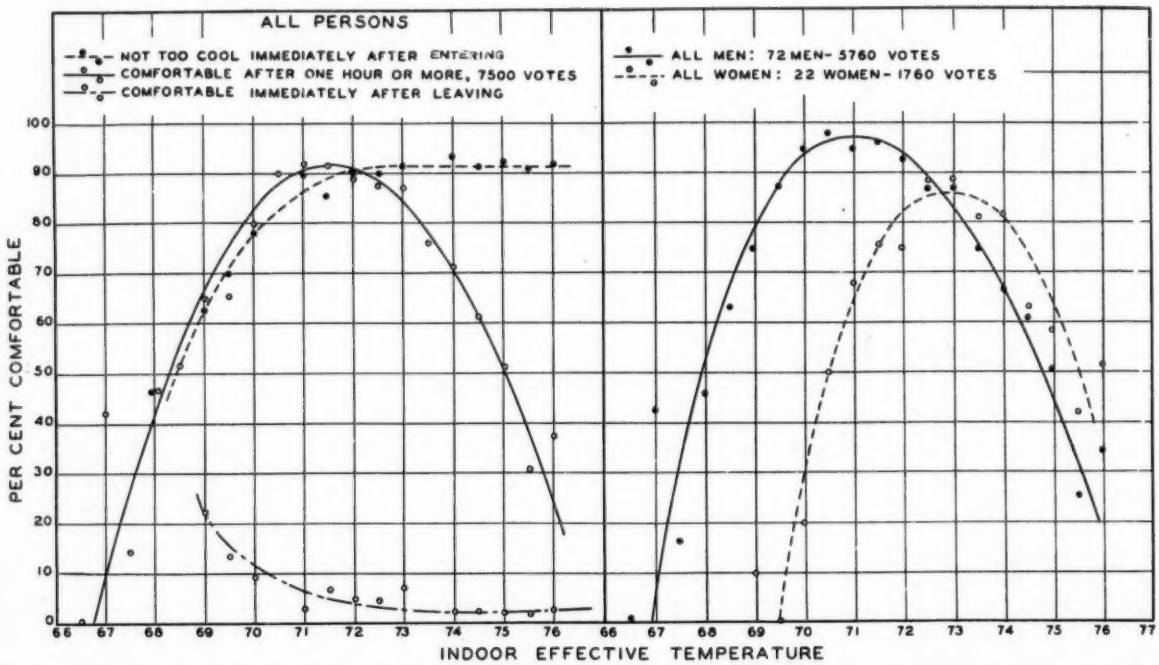
ALL PERSONS
NOT TOO COOL IMMEDIATELY AFTER ENTERING
COMFORTABLE AFTER ONE HOUR OR MORE, 7500 VOTES
COMFORTABLE IMMEDIATELY AFTER LEAVING
PER CENT COMFORTABLE
INDOOR EFFECTIVE TEMPERATURE
ALL MEN: 72 MEN- 5760 VOTES
ALL WOMEN: 22 WOMEN- 1760 VOTES

FIG. 6. RELATION BETWEEN EFFECTIVE TEMPERATURE MAINTAINED IN CONDITIONED SPACE AND PERCENTAGE OF OBSERVATIONS INDICATING COMFORT FOR ALL PERSONS

FIG. 7. RELATION BETWEEN EFFECTIVE TEMPERATURE MAINTAINED IN CONDITIONED SPACE AND PERCENTAGE OF ALL MEN AND OF ALL WOMEN INDICATING COMFORT

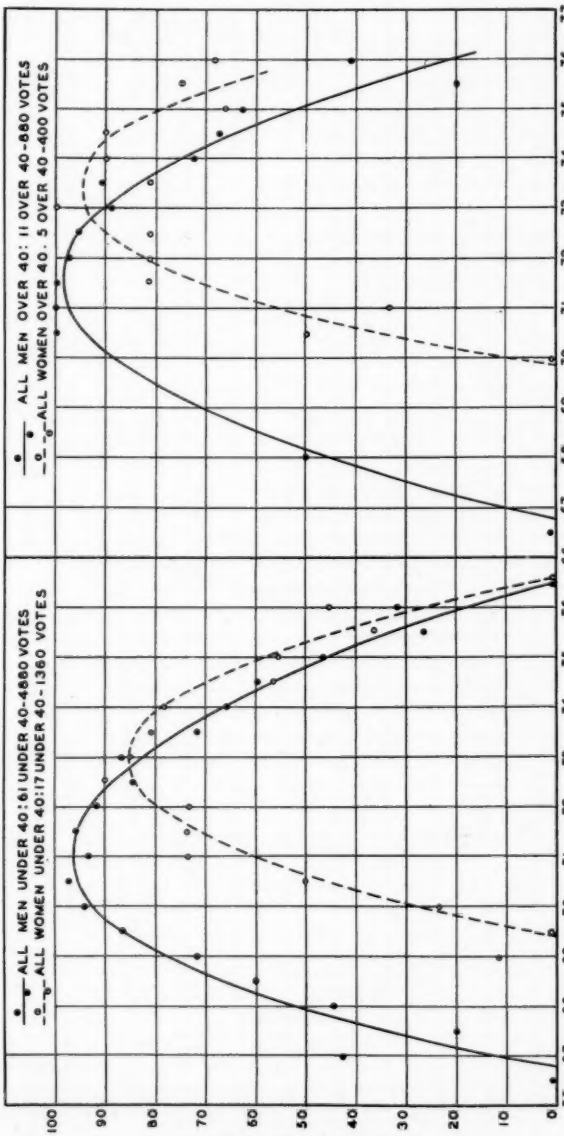

FIG. 8. RELATION BETWEEN THE EFFECTIVE TEMPERATURE MAINTAINED IN CONDITIONED SPACE AND THE PERCENTAGE OF OBSERVATIONS INDICATING COMFORT FOR ALL MEN AND FOR ALL WOMEN UNDER FORTY

FIG. 9. RELATION BETWEEN THE EFFECTIVE TEMPERATURE MAINTAINED IN CONDITIONED SPACE AND THE PERCENTAGE OF OBSERVATIONS INDICATING COMFORT FOR ALL MEN AND FOR ALL WOMEN OVER FORTY

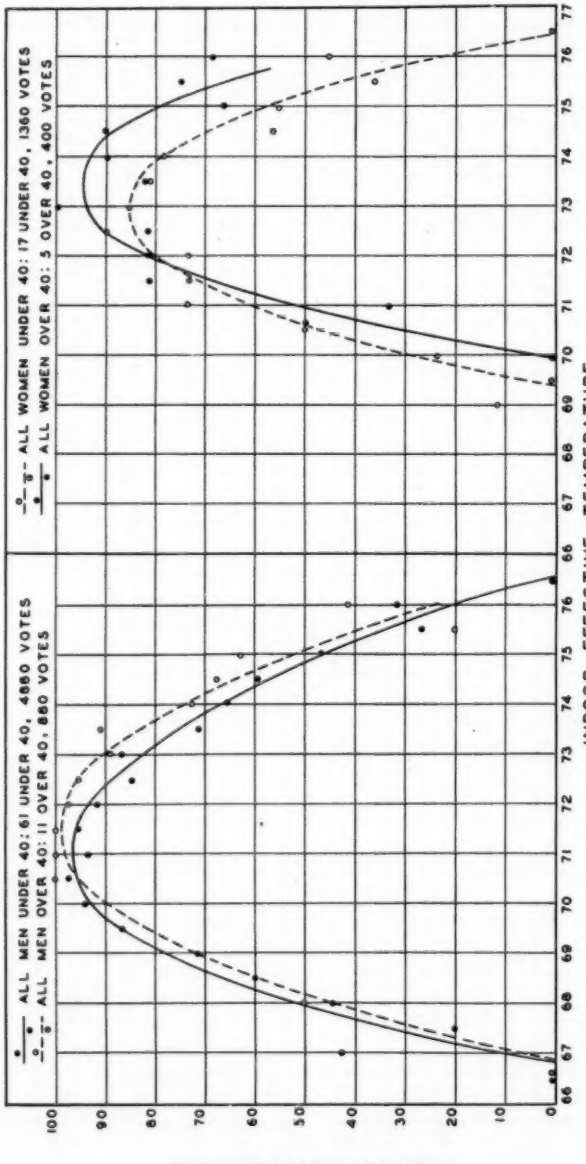

FIG. 10. RELATION BETWEEN THE EFFECTIVE TEMPERATURE MAINTAINED IN CONDITIONED SPACE AND THE PERCENTAGE OF OBSERVATIONS INDICATING COMFORT FOR ALL MEN UNDER FORTY AND ALL MEN OVER FORTY

FIG. 11. RELATION BETWEEN THE EFFECTIVE TEMPERATURE MAINTAINED IN CONDITIONED SPACE AND THE PERCENTAGE OF OBSERVATIONS INDICATING COMFORT FOR ALL WOMEN UNDER FORTY AND ALL WOMEN OVER FORTY

classed as perfect the optimum effective temperature seems to be about 1 deg higher than for those in perfect health.

There was no apparent difference in the reactions of slender men and stout men, but the comfort of the women was apparently dependent to some extent upon their size. The stout women indicated a desire for an effective temperature of 1 or 2 deg lower than the optimum for slender women.

It is known that the degree of activity of any individual has a definite effect on his comfort, but sufficient data were not obtained in this study to make a quantitative analysis of that effect.

A study of the reactions of all persons upon entering the air conditioned space, shown in Fig. 6, indicates that, regardless of the differential, there was very little shock felt when the inside effective temperature was not less than 71.5 F. With effective temperatures of less than 71.5 deg there was a noticeable shock, and as the effective temperature was lowered the percentage of persons who were too cool immediately after entering the building rose rapidly.

When the effective temperature was as high as 76 deg inside there were still a few persons who felt mildly cool on entering, and the percentage of mildly cool votes was about the same at 72 deg ET as it was at 76 deg ET; however, it is interesting to note that when the inside effective temperature was 72 deg, the percentage of persons who were not too cool immediately after entering (approximately 90 per cent) was practically the same as the percentage comfortable after an hour or more in the air conditioned space, whereas, when the effective temperature was 76 deg, the percentage of people comfortable after an hour dropped to approximately 31 per cent.

Each person was asked to indicate on his daily questionnaire (Fig. 3) the duration of his reaction immediately after entering and immediately after leaving the building. For the range of indoor effective temperatures observed, the majority indicated that they felt no shock on entering the building and that their feeling of comfort was no different after several hours in the conditioned space than immediately after entering. However, for those who did experience a cold shock on entering the building, the durations of the shock were as follows:

Not more than 5 min	29.6 per cent
Between 5 and 10 min	15.9 per cent
Between 10 and 15 min	16.9 per cent
Between 15 and 30 min	22.8 per cent
Between 30 min. and 1 hour	9.5 per cent
More than 1 hour	5.3 per cent

When the inside effective temperature was in the vicinity of the lower limit of the comfort zone, there was a noticeable lengthening of the duration of the cold shock on entering the building as shown by the curve in Fig. 12.

The warm shock that is felt by persons leaving an air conditioned building is a sensation that is not easily analyzed. It is needless to say that any condition is emphasized by contrast, but that should not necessarily mean that the condition is harmful because of the contrast. It is not reasonable to expect one to be comfortable in 98 F or 100 F air, even when the contrast between indoors and outdoors is negligible.

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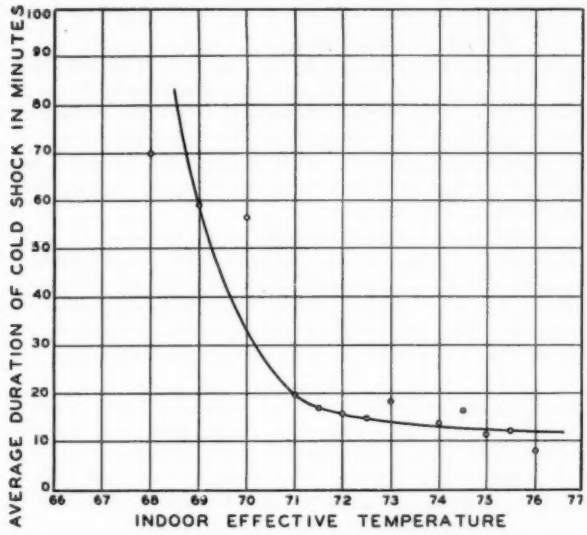


FIG. 12. RELATION BETWEEN THE EFFECTIVE TEMPERATURE MAINTAINED IN THE CONDITIONED SPACE AND THE DURATION OF COLD SHOCK ON ENTERING CONDITIONED SPACE

In this study an attempt was made to establish the relation between indoors and outdoors, but no definite relation was discernible. The warm shock seems to depend almost entirely on the outdoor temperature. The percentage of persons indicating a shock on leaving was practically the same when the dry-bulb temperature differential was 6 deg as when it was 18 deg.

Practically the entire group expressed a feeling of discomfort immediately after leaving the building. Each individual was asked to indicate the duration of any feeling of discomfort that was experienced on leaving. A number of them stated that they had a feeling of discomfort even after several hours. When a change was felt and the feeling of discomfort disappeared or was lessened, after an hour or more, it was probably due to the fact that the sun had gone down and the outdoor conditions were actually more comfortable. Therefore, in studying the duration of the warm shock, only those individuals were considered who recorded a duration of not more than one hour of discomfort after leaving the building. For that group, which was approximately 35 per cent of all subjects, the durations expressed were as follows:

Not more than 5 min	5.3 per cent
Between 5 and 10 min	5.8 per cent
Between 10 and 15 min	21.9 per cent
Between 15 and 30 min	28.4 per cent
Between 30 min and 1 hour	38.6 per cent

The 65 per cent of the subjects who stated that their feeling of discomfort lasted for more than 1 hour after leaving the conditioned space evidently did not consider their feeling of discomfort as a shock.

The percentage of all persons who were comfortable on leaving the conditioned space was noticeably higher when there were a number who felt mildly cool immediately before leaving as indicated in Fig. 6. In that case the contrast appears to have been pleasing at first rather than oppressive.

Near the end of the study when the employees had become more observant of their feelings of comfort, both in the office and at home, a questionnaire was given to each of them asking for their general reactions to air conditioning and their opinions on its over-all effectiveness. In order that everyone might give sufficient thought to this subject, the questionnaires were not collected until the second day after they had been distributed.

The following 3 questions given were asked in the questionnaire that was distributed during the last week of the study:

1. What difference, if any, have you noticed in your reactions this summer after leaving the office on warm days as compared to last summer previous to the installation of air conditioning?

2. Do you think that spending the day in an air conditioned office enables you to withstand the prolonged summer heat better than your friends and associates who are not in air conditioned offices during the day?

3. General remarks on the effectiveness of air conditioning.

Seventy-nine of these questionnaires were filled out and returned. Some of the answers to questions Nos. 1 and 2 were rather lengthy and somewhat vague, but, after each of them had been carefully studied for its exact meaning, and all had been classified, the results were as follows:

In answering question No. 1, the entire group apparently interpreted the phrase *after leaving the office* as meaning *immediately after leaving the office.* Fifty-six per cent stated that they were less comfortable this year than last year immediately after leaving the office; 14 per cent stated that they noticed no difference whatever; and 30 per cent stated that they were more comfortable this year than last year.

Of the answers to question No. 2, 86 per cent were affirmative and 14 per cent were negative.

Some of the answers to question No. 3 were meaningless and a few persons failed to answer it at all; however, most of the answers were very favorable. The percentages were as follows:

Favorable	67 per cent
Meaningless or no answer	23 per cent
Unfavorable	10 per cent

A few typical answers to question No. 3 are given:

I, personally, have developed sinus trouble. Whether it is directly due to air conditioning, I do not know.

I notice especially the quickness with which physical exertion tires and weakens me when I work or play outside in the summer heat.

I do not believe that air conditioning agrees with my health. At different times I have left the building and for 30 min or more have had a chill. At other times I felt very warm.

XUM

The temperature seems to be fine, but it is stuffy as if not enough fresh air were getting into the office.

I have not had my usual hay fever this year.

Air conditioning enables one to be comfortable and comfort is life's greatest joy.

It has helped my sinus trouble. It has made summer a pleasant time of the year instead of an ordeal that had to be endured. It has done away with paper weights. It has made it possible for one to maintain a uniform standard of work all the year around.

This is the most pleasant summer I have ever gone through.

A more courteous, cheerful attitude has been noticed in the office since the installation of air conditioning.

I think that the benefits have been very noticeable in myself and my fellow employees in that our energy does not appear to lag in the afternoons during a long, continued hot spell as it did before the installation of air conditioning.

I think that it makes you want to stay at your job in the office where it is cool and comfortable. I am more anxious to come to work since we have air conditioning.

It has been a pleasure to work in our building this summer. One great benefit has been in not having the clothing sticking to the body because of perspiration.

You feel as alive and fresh at the end of the day as you do in the morning.

Great!

The results of this phase of the study have further verified the generally accepted theory that the harmful effects of heat are cumulative, and that prolonged hot weather is not accompanied by the usual *run down* feeling when relief from the heat can be experienced through the hottest part of each day.

No. 1128

SEMI-ANNUAL MEETING, 1939

MEMBERS of the Society registered from 20 states in the United States and Canadian provinces at the Semi-Annual Meeting 1939, held in the Grand Hotel, Mackinac Island, Mich., July 4-6. Exceptional interest in the technical sessions was evident and those who attended count this Great Lakes Meeting outstanding in the history of Society meetings. Members and their families were delighted with the friendly spirit and the splendid hospitality of the host Chapters and greatly enjoyed the delightful program provided by the Committee on Arrangements.

Pres. J. F. McIntire, Detroit, called the first session of the Semi-Annual Meeting 1939 to order. O. D. Marshall, Grand Rapids, Chairman of Activities on Mackinac Island, welcomed the Society members on behalf of the Michigan and Western Michigan Chapters and briefly outlined the program that had been prepared for their enjoyment.

President McIntire responded on behalf of the members and thanked Mr. Marshall and the Committee on Arrangements for the splendid program that had been prepared. Mr. McIntire then gave a brief outline of the activities that had been undertaken by the Society since the Annual Meeting and gave a resumé of his observations while visiting local Chapters in the south, on the west coast, and in the middle west. He gave some statistics about the Society's financial condition, the election of new members, distribution of THE GUIDE, and commented on some of the important work assigned to special committees.

Messages of greeting from W. J. Olvany, New York, president of the *Heating, Piping and Air Conditioning Contractors National Association,* and A. W. Williams, Columbus, managing director of the *National Warm Air Heating and Air Conditioning Association,* were read.

The second session was called to order on Wednesday afternoon, July 5, and E. K. Campbell, Kansas City, Mo., chairman of the Membership Committee, was introduced and presented his report on the status of Society membership.

Report on Membership

Out in Kansas there is a group of country editors who have made quite a reputation for their wit and humor. One of them was asked why it was that a married man seemed to get further than a bachelor and his reply was, "No man runs his best unless there is someone after him."

I think it was on that theory that the Council appointed the Membership Com-

mittee with a Chairman whose business it is to keep after the Chapters and their Membership Chairmen, and it is not an easy job.

Giving the statistics on membership, we now have a total of 3127 members of all grades. During the first six months of 1939 we received 215 applications as compared to 237 applications in the first six months of 1938. During the same period 214 members have been elected, whereas 554 were elected in the same period of 1938. This large number in the first six months of 1938 was due to the fact that many applications were received during December 1937, because on the first of January the $10 initiation fee again became effective.

During the first six months of 1939 there was a loss of 190 members through resignation, cancellation or death; during the corresponding period in 1938 there was a loss of 193. During the second six months period of 1938 there were not so many, as only 20 were dropped. Hence, if we can keep the membership activity going during the balance of this year, we may be able to make a fair showing. Against the number dropped in the first six months of this year (190), 214 members have been actually elected, making a gain of 24. The Committee hopes for considerable more gain in the second six months.

There seem to be two theories or two schools of thought regarding the Membership Committee's activities. I have regarded the Chairmanship of the Membership Committee as that of a sales manager with a crew of salesmen, selected by someone else and over whom I have little or no authority, but I have regarded the work of obtaining new members strictly as a sales proposition. We have something to sell.

On the other hand, some Chapters and some members have considered that active attempts to "sell" membership in the Society have been beneath their dignity, and the dignity of the Society. This has had a very natural result in that the Chapter which particularly upholds that idea has the smallest membership in proportion to population of all of the Chapters in the Society.

In my judgment it makes no difference how good a thing a person has to sell or give away, if he does not let anyone know about it he is neither going to sell nor give away very much. The world makes a beaten path to the door of the best mouse trap, only when the world knows about it. If we wait in our membership work for someone else to spread the news, we are not going to sell or give away or share the privilege of membership in this Society to any great extent.

I regard it as a business of the Membership Committee and of the Chapter Membership Committees to spread the news, to sell membership. This Society is doing a great work, and I know of no other similar Society doing as good work, and as important work as our Research. Our Research program has grown until, including both sides of our cooperative programs, it represents an expenditure in the neighborhood of $100,000 annually. Our research work has developed many facts that were in the cloudy area when we began this work. The results of its work have been far reaching.

So we have something to sell and we have something to sell that is worth buying on the part of prospective customers. We are giving value received whenever we sell a membership, and any man engaged in any of the allied industries, receiving as he does material benefits from this Society, should be willing to become a member if for no other reason than that he is willing to help.

Having this thought in mind, you will find, as I have found, that many times it is necessary only to suggest to someone connected with the industry or the profession that it would be to his interest to become a member, and that is all that is required. If each member of the Society (and I wish you would take this message back to your Chapters) will simply carry it in his mind to give men of the right sort an opportunity to become members, you will be surprised at the number of new members who will join.

E. K. CAMPBELL, *Chairman*,
MEMBERSHIP COMMITTEE

The final session of the Semi-Annual Meeting, 1939 was opened on Thursday morning, July 6, by Pres. J. F. McIntire who introduced F. C. Houghten, director of the Society's Research Laboratory, Pittsburgh, and W. L. Fleisher, New York, chairman of the Committee on Research, who

summarized the report of the committee. He referred to several outstanding accomplishments during the first six months of 1939, one being the consummation of an agreement with the U. S. Navy Department, Washington, D. C., which involves a comprehensive cooperative program extending over a period of two years. The program is under the direction of a special committee composed of W. L. Fleisher, J. H. Walker, C.-E. A. Winslow, A. E. Stacey, Jr., and Director F. C. Houghten.

Research Projects at Pittsburgh

(1) From a contribution of $3,500 in earmarked funds, test cubicles to study the solar radiation transmittance through glass block wall construction have been erected at the Laboratory. Because this study is of vital interest to engineers concerned with the application of summer cooling and air conditioning in buildings containing glass block construction, the results will be of wide interest. The work is proceeding under the supervision of the Technical Advisory Committee on Air Conditioning Requirements of Glass, of which M. L. Carr is chairman.

(2) A fund of $6,000 has been collected for studies outlined by the Technical Advisory Committee on Radiation and Comfort, of which J. C. Fitts is chairman. Two test rooms have been erected on the roof of one of the Laboratory buildings in Pittsburgh and arranged with similar exposures in order that the effect of radiant heat may be appraised on a comparative basis with other commonly accepted methods of heating.

(3) Studies of the flow of air in ducts continue under the direction of Chairman J. H. Van Alsburg and his Technical Advisory Committee on Air Distribution and Air Friction. The present work involves round elbows and their square equivalent after which investigations will be conducted on straight rectangular ducts.

(4) Chairman C. S. Leopold and his Technical Advisory Committee on Cooling Load in Summer Air Conditioning have been instrumental in securing funds for the erection of two test buildings to determine the heat flow through various built-up wall sections.

(5) The physiological reactions of subjects to various heat outputs in summer and winter are being studied by the Technical Advisory Committee on Air Conditioning in Industry, of which A. E. Stacey, Jr., is chairman.

(6) Data analyzed by the Laboratory staff for summer cooling requirements in different metropolitan districts are presented in a paper at this meeting. Under the direction of C. Tasker and his Technical Advisory Committee on Sensations of Comfort, approval has been given for studies outlined by a Joint Committee of the ASHVE and the *Illuminating Engineering Society* which will be conducted at the Laboratory to determine human reactions to color sensations in various environmental conditions.

Cooperating Institutions

Research under cooperative agreements is being conducted at the following colleges:

University of California. Project outlined by the Technical Advisory Committee on Cooling Towers, Evaporative Condensers and Spray Ponds, of which B. M. Woods is chairman.

Case School of Applied Science. Two projects dealing with air distribution are under investigation. Another study comes under the direction of the Technical Advisory Committee on Heat Transfer of Finned Tubes with Forced Air Circulation, of which G. L. Tuve is chairman.

University of Illinois (*Engineering—Urbana*). This investigation is concerned with the effect of inlet and exhaust opening location on the motion and distribution of air in a room.

University of Illinois (*Medical—Chicago*). Data are being collected on the deter-

mination of cardiac output, blood volume, skin temperatures and oxygen saturation of venous blood at comfortable, hot, dry and wet conditions for various subjects.

Lehigh University. Commencing July 1 an investigation started on round and square elbows to determine resistance offered to air flow.

Massachusetts Institute of Technology. Studies are planned at this institution on the attenuation of sound through lined and unlined ducts. This is one phase of the program of the Technical Advisory Committee on Sound Control, J. S. Parkinson, chairman.

Michigan College of Mining and Technology. Through the Technical Advisory Committee on Corrosion a program has been developed at this institution under the direction of A. R. Mumford, who is chairman of this committee dealing with the determination of the degree of contamination of condensate formed at varying rates from steam of known composition.

University of Minnesota. Investigation of air filter performance and methods of rating air filters is being continued under a renewal agreement effective July 1. A suitable field test is also being studied.

University of Pennsylvania. This research contemplates an experimental investigation of the departures from Dalton's law in the case of moist air and has been planned by the Technical Advisory Committee on Psychrometry of which F. R. Bichowsky is chairman.

University of Pittsburgh. Studies at the Elizabeth Steel Magee Hospital deal solely with the bacteriological content of the air in an air conditioned operating room and have been supervised by the Technical Advisory Committee on Treatment of Disease, of which Dr. M. B. Ferderber is chairman.

Agricultural and Mechanical College of Texas. This research under the direction of the Technical Advisory Committee on Heat Requirements of Buildings, of which P. D. Close is chairman, is intended to ascertain flow of heat through walls for both heating and cooling cycles.

University of Wisconsin. This is another study which comes under the supervision of the Technical Advisory Committee on Air Distribution and Air Friction.

Advisory Committees' Activities

Committee on Air Pollution and Purification. Realizing that the membership of the Society has a definite interest in the rather broad field of air pollution and purification the Research Executive Committee approved the appointment of C.-E. A. Winslow to act as general chairman for the ensuing year. An invitation was extended to H. B. Meller to act as chairman for the Sub-Committee on Air Pollution and Dr. Leonard Greenburg has accepted the invitation to become chairman of the Sub-Group on Removal of Atmospheric Impurities. The personnel of this general committee has just been completed and it is expected that in the near future a program of activity will be outlined and initiated.

Committee on Instruments. This committee was formed with D. W. Nelson, chairman, and its program will be announced shortly.

Committee on Weather Design Conditions. Chairman T. H. Urdahl has been setting up a program in cooperation with the various utility organizations to determine uniformly outdoor dry- and wet-bulb temperatures in the principal cities.

Committee on Solid Fuels. Chairman W. A. Danielson has designated several sub-groups to report on anthracite, bituminous and sub-bituminous coals as well as lignite and coke. Another sub-committee on chimneys has been very active since the first of the year in outlining a program of research.

Committee on Insulation. R. T. Miller, chairman, reports cooperative work under way with the Guide Publication Committee and development of a chart showing the recommended relative humidities to be maintained inside the buildings with varying outdoor conditions.

Vice-Pres. F. E. Giesecke, College Station, Tex., took the chair and expressed the appreciation of the members for the papers and inquired if there was any unfinished business to come before the Society.

As nothing was offered, he called for the report of the Resolutions Committee, submitted by R. H. Carpenter, *chairman;* H. H. Erickson and W. A. Russell, which was read by Secy. A. V. Hutchinson, New York, and was unanimously adopted.

Report of Resolutions Committee

The members of the Society attending the Semi-Annual Meeting 1939 at Mackinac Island, July 4, 5 and 6, having enjoyed to the utmost the thoughtful and complete arrangements which have contributed so much to the delightful meeting now coming to a close, wish to express their thanks and appreciation as follows:

To W. G. Boales, general chairman, and his able and hard working Committee on Arrangements of the Michigan and Western Michigan Chapters for their untiring efforts in our behalf;

To the membership of the two Chapters as a whole for their splendid cooperation in their joint sponsorship of the meeting;

To Emery Hatch and staff of the Georgian Bay Line for the courteous assistance rendered those members who came by boat;

To W. S. Woodfill and every employee of The Grand Hotel whose unfailing thoughtfulness and courtesy have contributed so largely to our pleasure;

To Col. Roger Andrews, whose all too brief talk on this historic spot at the semi-annual banquet was outstanding. It was not only a highlight of our meeting, but it also aroused in all of us a desire for further knowledge about this Hill of History;

To the trade press and newspapers who devoted so much space to recording the events of this meeting and to the *Mackinac Island News* for the gracious write-up in the July 1 issue, and finally,

To the authors of the papers presented at this meeting for their patience in collecting the necessary data and the time required to present it in proper form;

Be It Therefore Resolved, That the above is the sentiment of this Semi-Annual Meeting and that a copy of these resolutions be sent to each of the above named individuals and organizations.

President McIntire assumed the chair and made the concluding announcements, after which the Semi-Annual Meeting 1939 adjourned.

Council Meetings

At Mackinac Island on July 4 and 6, the Council held two meetings.

A petition from Portland, Oregon, members requesting a charter for a Chapter in Portland was approved.

Nomination of five members to serve on the Committee on Research for a three-year term beginning 1940 was announced as follows: Philip Drinker, Boston, Mass.; Axel Marin, Ann Arbor, Mich.; A. E. Stacey, Jr., New York, N. Y.; J. H. Van Alsburg, Chicago, Ill.; J. H. Walker, Detroit, Mich.

Research Committee

The Committee on Research met at Mackinac Island on July 4 with 12 members and several officers of the Society and guests in attendance. Cooperative agreements were extended with the University of Wisconsin and Case

School of Applied Science for investigations on air distribution and with the University of Minnesota for a program of air filter research to be approved by the Technical Advisory Committee on Air Pollution and Purification.

Several reports were received, including that of the Finance Committee, on the possibility of cooperation with the U. S. Bureau of Standards, and the Air Conditioning Committee of Edison Electric Institute. Director F. C. Houghten reviewed a comprehensive outline of Society research, which he had prepared, and summarized projects at the Laboratory in Pittsburgh.

Discussion was given to the Proposed Regulations Governing ASHVE Research Technical Advisory Committees, which are to be sent to members of the Committee on Research for approval by letter ballot.

PROGRAM SEMI-ANNUAL MEETING

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
THE GRAND HOTEL, MACKINAC ISLAND, MICH.
JULY 4-6, 1939

July 4

8:30 A.M. Registration—Peacock Alley—(Office Floor)
10:00 A.M. Council Meeting (Crystal Room)
2:30 P.M. Meeting Committee on Research (Crystal Room)

July 5

9:00 A.M. Committee on Air Distribution and Air Friction, Room 32 (Office Floor)
9:00 A.M. Technical Session—(Blue Room)
Greetings by O. D. Marshall
Response by Pres. J. F. McIntire
Review of Society Activities by President McIntire
The Effect of Vibration upon Free Convection from Horizontal Cylinders by R. C. Martinelli and L. M. K. Boelter
Heating Low Cost Homes by R. K. Thulman and L. E. Seeley
Evaluating Performance of Water Heaters Fired with Solid Fuels by H. J. Rose and R. C. Johnson
11:00 A.M. Committee on Radiation and Comfort, Room 30
12:00 NOON Committee on Sensations of Comfort
12:00 NOON Meeting—Nominating Committee (Luncheon Crystal Room)
1:30 P.M. Technical Session (Blue Room)
Report of Membership Committee—E. K. Campbell
Ozone in Ventilation—Its Possibilities and Limitations by W. N. Witheridge and C. P. Yaglou
Effect of Size and Type of Air Inlet and Outlet on the Heat Output of Convectors by A. P. Kratz, M. K. Fahnestock and E. L. Broderick
A Theory Covering the Transfer of Vapor through Materials by F. B. Rowley
Fire Protection for Air Conditioning Systems by R. C. Loughead

July 6

9:00 A.M. Technical Session (Blue Room)
Report of Committee on Research by W. L. Fleisher
Summer Cooling Requirements in Washington, D. C. and Other Metropolitan Districts by F. C. Houghten, Carl Gutberlet and Albert A. Rosenberg.
Study of Summer Cooling in the Research Residence for the Summer of 1938 by A. P. Kratz, S. Konzo, M. K. Fahnestock and E. L. Broderick
Cooling Tower Performance Studies by L. M. K. Boelter
Unfinished Business — New Business — Resolutions — Adjournment

ENTERTAINMENT

July 4

9:30 A.M. Golf Tournament (Greens Fee $2 per day) Hotel Course and Country Club Course
18-hole Medal Play—Prizes for Members and Guests. Register with Golf Committee and furnish handicap information. Hotel course reserved exclusively for ASHVE tournament players until 3:00 p.m., Tuesday, July 4. Participants in Research Cup and Chapter Tournament must play hotel course on Tuesday. Players in Kicker's Handicap may play either hotel or country club course.
10:30 A.M. Sports Program and Children's Games. Register for tennis, badminton, archery, quoits, croquet
11:00 A.M. Lake Trout Trolling Trip
Register with S. H. Downs. Boat leaves Union Dock. Fee $5 per person includes tackle, bait, services of guide, luncheon and refreshments. Return at 4:00 p.m.
1:30 P.M. Golf Tournament—no charge for transportation between courses for tournament players.
3:00 P.M. Children's Hiking Party
3:00 P.M. Baseball game between local teams (Grand Hotel Ball Field)
3:30 P.M. Water Sports at Pool (fee for bathers 75c) (Children 25c)
5.45 P.M. Children's Bus Ride
6:00 P.M. Social Hour (Dutch Treat Club $1 per person—Blue Room)
9:00 P.M. Civic Celebration—Folk Dancing
10:00 P.M. Dance and Floor Show (Casino)

July 5

10:00 A.M. Children's Sports
10:00 A.M. Bicycle Party (35c per hour bicycle rental)
11:00 A.M. Carriage Ride Around the Island ($1.50 per person) (Children 5-12 75c)
2:15 P.M. Boat Ride for Ladies and Children. Steamer Mackinac Islander leaves Union Dock
4:30 P.M. Tea Dance and Concert in Tea Garden
6:00 P.M. Children's Beach Party and Picnic Supper
7:00 P.M. Semi-Annual Banquet and Dance
Prof. E. O. Eastwood, Toastmaster. Presentation of Golf Cups. Presentation of Past President's Memory Book to E. Holt Gurney. Address, The Hill of History, by Col. Roger Andrews.

July 6

10:30 A.M. Hiking Party to Fort Mackinac and Visit to State Museum
1:30 P.M. Informal Golf Matches (Greens Fee $1.75 per day)
1:30 P.M. Ladies Bridge
2:00 P.M. Horseback Riding ($1.50 per hour)
Boardwalk stroll and Ladies' Shopping Tour—Visit to Yacht Harbor and Astor House

COMMITTEE ON ARRANGEMENTS

W. G. BOALES, *General Chairman*, Detroit

S. H. Downs, Kalamazoo
C. R. McConner, Kalamazoo
B. F. McLouth, Lansing
F. J. Linsenmeyer, Detroit
W. C. Randall, Detroit
J. H. Walker, Detroit

Finance: J. S. Kilner, Detroit; C. R. McConner, Kalamazoo; F. J. Linsenmeyer, Detroit.

Attendance: F. C. Warren, Grand Rapids, *Chairman;* C. H. Morton, Grand Rapids; L. L. McConachie, Detroit; E. M. Harrigan, Detroit.

GREETERS COMMITTEES

Detroit: G. H. Tuttle, *Chairman;* G. W. Akers, E. H. Clark, W. C. Randall, F. J. Feely, J. S. O'Gorman, M. B. Shea.

Cleveland: D. L. Taze, *Chairman;* L. T. Avery, W. R. Rhoton.

Buffalo: J. J. Landers, *Chairman;* M. C. Beman, Joseph Davis, C. W. Farrar, L. P. Saunders.

Chicago: Tom Brown, *Chairman;* J. R. Vernon.

S.S. North American: R. E. Moore, Chicago; A. P. Kratz, Urbana; J. J. Aeberly, Chicago.

S.S. South American: F. J. Feely, Detroit; R. K. Milward, Detroit; R. F. Connell, Detroit; W. J. Whelan, Detroit; G. H. Tuttle, Detroit; F. W. Johnson, Detroit.

Transportation in Detroit: M. B. Shea, *Chairman;* J. F. Lance, L. A. Burch.

Bon Voyage Committee, Detroit: E. H. Clark, *Chairman;* M. B. Shea, L. A. Burch, C. L. Toonder.

Canadian Trips: W. J. Whelan, Detroit; R. K. Milward, Detroit; F. J. Linsenmeyer, Detroit; E. J. Anderson, Detroit; R. L. Deppmann, Detroit.

COMMITTEES FOR MACKINAC ISLAND ACTIVITIES

O. D. MARSHALL, Grand Rapids, *Chairman.*

Reception: R. F. Connell, Detroit, *Chairman;* J. S. O'Gorman, Detroit; M. W. Bishop, Chicago; F. C. Warren, Grand Rapids; G. H. Tuttle, Detroit; S. W. Todd, Grand Rapids; S. H. Downs, Kalamazoo.

Transportation: H. F. Reid, Muskegon Heights, *Chairman;* S. S. Sanford, Detroit, *Vice-Chairman;* J. S. Cunningham, Dowagiac; W. H. Old, Detroit; H. D. Bratt, Grand Rapids; W. F. Graff, Grand Rapids; J. G. Dykman, Grand Rapids; P. J. Vanderlip, Lansing; H. J. Young, Muskegon; S. J. Dempsey, Battle Creek.

Publicity: T. D. Stafford, Grand Rapids, *Chairman;* S. L. Worthing, Grand Rapids; H. F. Hutzel, Detroit.

Golf: G. D. Winans, Detroit, *Chairman;* W. G. Schlichting, Kalamazoo; A. E. Knapp, Detroit; R. L. Deppman, Detroit; E. J. Anderson, Detroit.

Children's Activities: S. S. Sanford, Detroit, *Chairman;* Mrs. F. J. Feely, Mrs. J. H. Walker, Mrs. A. E. Knapp, Mrs. O. D. Marshall, Mrs. R. F. Connell, Mrs. G. D. Winans, Mrs. S. H. Downs, Mrs. G. H. Tuttle.

Sports: B. F. McLouth, *Chairman,* and Mrs. McLouth, Lansing; Mr. and Mrs. R. E. Distel, Lansing; Mr. and Mrs. V. H. Hill, Lansing.

Fishing Trip: S. H. Downs, *Chairman,* Kalamazoo; H. J. Metzger, Kalamazoo; T. D. Stafford, Grand Rapids.

Sessions: W. W. Bradfield, Grand Rapids, *Chairman;* W. C. Randall, Detroit, *Vice-Chairman;* R. S. M. Wilde, Highland Park; A. E. Knibb, Detroit; G. D. Woodhouse, Dowagiac.

Ladies: H. F. Hutzel, Detroit, *Chairman;* B. F. McLouth, Lansing; Mrs. L. G. Miller, East Lansing; Mrs. C. R. McConner, Kalamazoo; Mrs. L. L. McConachie, Detroit; Mrs. O. D. Marshall, Grand Rapids; Mrs. R. F. Connell, Detroit; Mrs. Tom Brown, Chicago.

Special Boat Trips: C. H. Pesterfield, East Lansing, *Chairman;* E. J. Anderson, Detroit; B. F. McLouth, Lansing; S. S. Sanford, Detroit; R. K. Milward, Detroit.

Banquet: L. G. Miller, East Lansing, *Chairman;* F. R. Bishop, Detroit; A. Marin, Ann Arbor; W. C. Randall, Detroit.

No. 1129

THE EFFECT OF VIBRATION UPON FREE CONVECTION FROM HORIZONTAL CYLINDERS

By R. C. MARTINELLI,* MARTINEZ, CALIF., AND L. M. K. BOELTER,** BERKELEY, CALIF.

THE rate of heat transfer from a 0.75 in. diameter horizontal cylinder, being vibrated sinusoidally as a whole in a vertical direction, to surrounding water has been determined experimentally. The amplitude of vibration of the cylinder was varied from zero to 0.10 in. and the frequency of vibration from zero to 40 cycles per second. The difference in temperature between the cylinder and the surrounding fluid was varied from 8 F to 45 F.

The experimental results have been correlated using the dimensionless parameters, Nusselt (Nu), Grashof (Gr), Prandtl (Pr), and Reynolds (Re). In the Reynolds number used in the correlation the velocity term represents the root mean square velocity of the vibrating cylinder.

Velocities of vibration yielding magnitudes of Reynolds' Modulus less than a *critical value* (about 1000), have no effect on the rate of heat transfer from the cylinder within the range of experimental data, ($5 \times 10^5 \leq \mathrm{Gr \cdot Pr} \leq 5 \times 10^6$). At these low velocities of vibration the heat loss from the cylinder is largely due to free convection forces; that is, the difference in temperature between the cylinder and the ambient fluid determines the rate of heat transfer.

As Reynolds' Modulus (velocity of vibration) is increased beyond this critical value, the rate of heat transfer increases with an increased velocity of vibration. At a sufficiently high magnitude of Reynolds' Modulus, the effect of free convection becomes negligible and the forced convection due to the vibration controls.

Extrapolation of the data beyond the experimental limits for each variable, or their application to other boundary conditions, is not recommended.

APPARATUS

The apparatus utilized to measure the rate of heat transfer consisted of a horizontal cylinder 0.75 in. in diameter and 12⅝ in. effective length which

* Engineer, Shell Oil Co., Martinez, Calif.
** Professor, Mechanical Engineering, University of California, Berkeley, Calif.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Mackinac Island, Mich., July, 1939.

was vibrated as a whole in a vertical direction (normal to the cylinder axis), by means of a connecting rod and crank mechanism. An electric heater and thermocouples were enclosed in the cylinder.

The cylinder was placed in the center of a 36×36 in. cylindrical tank filled with water. A photograph of the experimental station is shown in Fig. 1, and views of the cylinder with connecting rod and vibrating mechanism are illustrated in Fig. 2.

The temperature of the cylinder, the temperature of the ambient water, the power input to the heater, the frequency of vibration and the amplitude of the vibration were measured during each run.[1]

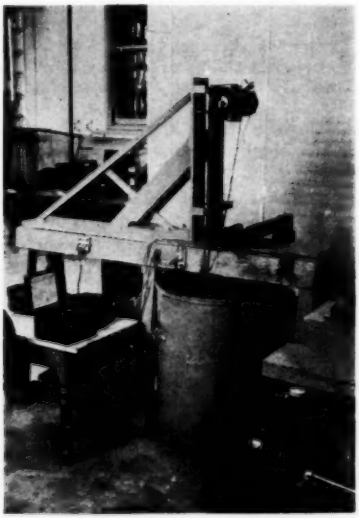


FIG. 1. ARRANGEMENT OF APPARATUS

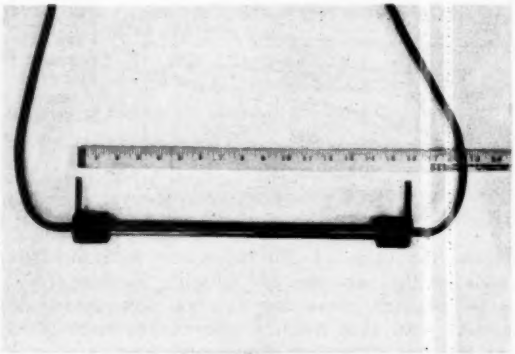
The amplitude and frequency of vibration were kept constant for four runs, while the power input to the heater was varied in order to note the effect of the free convection modulus (Grashof's Modulus) on the rate of heat transfer.

DISCUSSION OF RESULTS

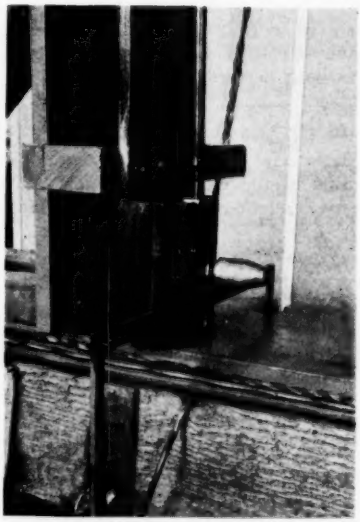
The results of the tests are shown plotted in Figs. 3 and 4. Fig. 3 reveals the boundary modulus Nu (proportional to the film transfer factor from the

[1] The Effect of Vibration on Heat Transfer by Free Convection from a Horizontal Cylinder, by R. C. Martinelli, and L. M. K. Boelter. (*Proceedings of the Fifth International Congress of Applied Mechanics*, Cambridge, Massachusetts, 1938, p. 578. John Wiley & Sons, 1939.)

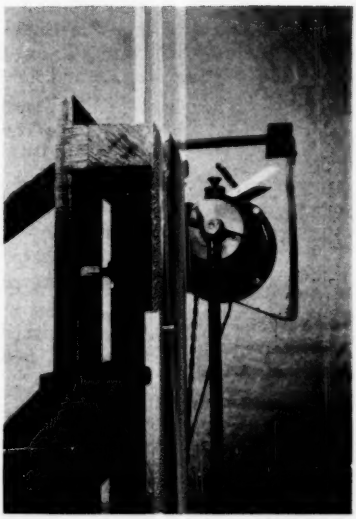
tube to the water) as a function of the free convection modulus times the fluid property modulus (Gr·Pr) for various magnitudes of Reynolds' Modulus (Re). It is noted that for no vibration and for small values of Re (less



a. HORIZONTAL CYLINDER



b. CYLINDER SUPPORT



c. CONNECTING ROD AND CRANK

FIG. 2. VIEW OF CYLINDER WITH CONNECTING ROD AND VIBRATING MECHANISM

than 1000) the data obtained check well with the free convection data for stationary tubes [2] obtained by others. As Reynolds' Modulus (Re) is increased beyond 1000, a new curve of Nu *vs* Gr·Pr is obtained for each

[2] Heat Transmission, W. H. McAdams. (McGraw Hill Co.)

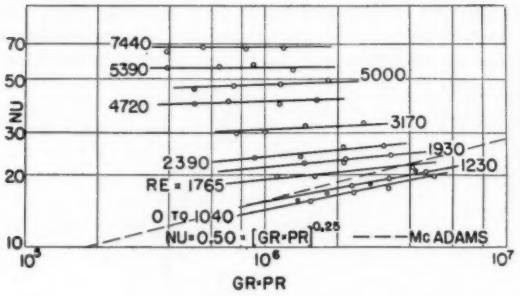


FIG. 3. VARIATION OF BOUNDARY MODULUS (NU) WITH PRODUCT OF FREE CONVECTION MODULUS AND FLUID PROPERTY MODULUS (GR · PR) FOR VARIOUS MAGNITUDES OF REYNOLDS' MODULUS (RE)

magnitude of Re, the value of Nu increasing with increasing magnitudes of Re. In addition the variation of Nusselt's Modulus (Nu) with Gr·Pr (proportional to the free convection forces) decreases as the velocity of vibration increases until at a magnitude of Re equal to about 7000, Nu is independent of the free convection forces and depends upon Re only.

This result follows because the role of the free convection heat loss becomes of smaller importance as the vibrational velocity is increased.

Cross plots of the curves in Fig. 3 are shown in Fig. 4 and illustrate clearly the variation of Nu with Re for constant values of Gr·Pr. An algebraic expression relating Nu, Re and Gr·Pr has been derived semi-analytically:

$$\mathrm{Re} = \mathrm{Nu}\sqrt{12{,}000 - 20\,\frac{(\mathrm{Gr}\cdot\mathrm{Pr})^2}{\mathrm{Nu}^8}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

As seen in Fig. 4 the expression conforms to the experimental data quite well. Complete details of the derivation of Equation (1) will be found in reference 1.

SEVERAL EXAMPLES OF APPLICATION OF RESULTS

A tube ½ in. in diameter carrying steam is being used to heat water to 150 F. The steam temperature is 250 F. What amplitude of vibration (of

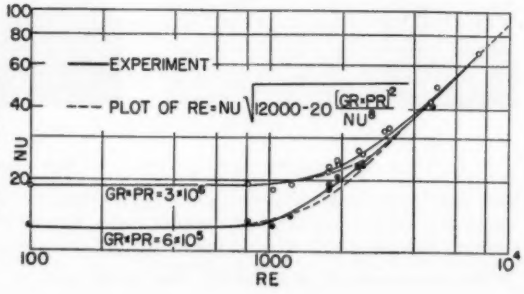


FIG. 4. BOUNDARY MODULUS (NU) AS A FUNCTION OF REYNOLDS' MODULUS (RE) FOR TWO MAGNITUDES OF (GR · PR). EXPERIMENTAL POINTS AND THE GRAPHICAL REPRESENTATION OF THE EMPIRICAL EQUATION

the tube) is required to double the heat transfer rate if the frequency of vibration is 30 cycles per second?

1. Computation of the thermal conductance per unit area for stationary tubes. Reference 2, page 249, indicates $Nu = 0.525\ (Gr \cdot Pr)^{1/4}$, an empirical relation between Nu and Gr·Pr for stationary tubes applicable in the range $10^3 \leq Gr \cdot Pr \leq 10^7$. The product Gr·Pr may be rewritten as follows:

$$Gr \cdot Pr = \alpha d^3 \Delta t, \text{ where } \alpha = \frac{\rho^2 g^2 \beta C_p}{3600 \mu k}\ (\text{ft})^{-3}\ (\text{F})^{-1}$$

Magnitudes of α for water and air are obtained from the literature:

$\alpha = 18.6 \times 10^8\ (\text{ft})^{-3}\ (\text{F})^{-1}$ at an average temperature of 200 F
$d = 1/2 \times 1/12 = 4.16 \times 10^{-2}$ ft
$\Delta t = 250\ \text{F} - 150\ \text{F} = 100\ \text{F}$
$\therefore\ Gr \cdot Pr = 18.6 \times 10^8 \times (4.16)^{-3} \times 10^{-6} \times 100$
$= 13.4 \times 10^6$
Nu for stationary tubes
$Nu = 0.525\ (13.4 \times 10^6)^{1/4} = 31.8.$

To double the film conductance per unit area Nusselt's Modulus (Nu) must be doubled. Thus, the magnitude of Nu required $= 31.8 \times 2 = 63.6$. Substituting the magnitudes of Nu and Gr·Pr into equation (1) yields

$$Re = 63.5 \sqrt{12{,}000 - 20\left[\frac{13.4 \times 10^6}{(63.6)^4}\right]^2}$$

$$= 6950 = \frac{a\omega d\rho}{\sqrt{2}\,\mu}$$

However:

$\omega = 30 \times 2\pi = 189$ radians per second
$d = 1/2 \times 1/12 = 4.16 \times 10^{-2}$ ft
$\mu/\rho = 0.34 \times 10^{-5}$ ft² per second for water at 200 F

$$\therefore a = \frac{1.414 \times 0.34 \times 10^{-5} \times 6950}{189 \times 4.16 \times 10^{-2}}$$

$= 4.23 \times 10^{-3}$ ft
= 0.051 in., the amplitude required to double the heat transfer rate at 30 cycles per second. The total displacement of the tube is twice this magnitude, that is, 0.102 in.

If a body is losing heat in air, the vibration is not as effective in increasing the rate of heat transfer as it would be if water is the ambient fluid. This fact may be shown by comparing the magnitudes of the velocities of vibration which yield the same value of Reynolds' Modulus (Re) in air and in water.

$$\text{Let } \left(\frac{a\omega d\rho}{\sqrt{2}\,\mu}\right)_{\text{air}} = \left(\frac{a\omega d\rho}{\sqrt{2}\,\mu}\right)_{\text{water}}.$$

For the same tube diameter,

$$d_{\text{air}} = d_{\text{water}};\ \text{then } \frac{(a\omega)\ \text{air}}{(a\omega)\ \text{water}} = \frac{(\rho/\mu)\ \text{water}}{(\rho/\mu)\ \text{air}}$$

$$(\rho/\mu)\ \text{water} = 2.93 \times 10^5\ \frac{\text{second}}{\text{ft}^2}\ \text{at 200 F}$$

$$(\rho/\mu)\ \text{air} = 4.17 \times 10^3\ \frac{\text{second}}{\text{ft}^2}\ \text{at 200 F}$$

The ratio of the vibrational velocity required in air to that required in water to give the same Re is 70.3:1. It follows that vibrating a body in air

will have no effect on the rate of heat transfer, until it is vibrated about 70 times as fast as would be necessary in water.

The purpose of this paper has been to present quantitatively the effect of vibration on free convection. The effect of vibration in air at frequencies and displacement amplitudes normally encountered is negligible. However, in water the effect becomes appreciable, the increase over free convection heat transfer being four fold for Reynolds' Moduli equal to 10,000. For Reynolds' Modulus less than 1000 no appreciable effect of vibration is observed. The minimum magnitude of Reynolds' Modulus at which vibration becomes effective increases with an increase in free convection rate of heat transfer. For heat transfer to fluids similar to water, in apparatus subjected to vibration, a liberal design results if the apparatus behavior prediction is based upon data obtained from stationary equipment.

ACKNOWLEDGMENTS

The authors wish to acknowledge their indebtedness to Wilson Tripp, who paved the way for this work; to C. J. Carlson, who worked with the authors at the beginning of the investigation; and to S. Polland and H. Johnson who obtained some of the experimental data. Technological assistance was had through WPA project No. 6090-5970

TABLE OF NOMENCLATURE

a = amplitude of displacement of the tube	feet
c_p = specific heat of fluid	Btu per (pound) (F)
d = diameter of tube	feet
f = thermal conductance per unit area	Btu per (foot)^2 (hour) (F)
g = gravitational acceleration = 32.2 ft per second^2	
k = thermal conductivity of the fluid	Btu per (foot)^2 (hour) (F per foot)
t = temperature	F
t_p = temperature of pipe	F
t_w = temperature of the ambient fluid	F
$\triangle t = t_p - t_w$	F
$\alpha = \frac{\rho^2 g^2 \beta C_p}{3600 \mu k}$ = property of fluid	feet$^{-3} \cdot$ (F)$^{-1}$
β = expansion coefficient of fluid	(F)$^{-1}$
γ = weight density of fluid	pound per foot^3
μ = viscosity of fluid	(pound)(second) per foot^2
ν = kinematic viscosity of fluid	feet^2 per second
ω = angular velocity of vibration	second^{-1}
$\omega/2\pi$ = frequency of vibration	second^{-1}
ρ = mass density of fluid	pound second^2 per foot^4

All properties are evaluated at $(t_p + t_w)/2$.

DIMENSIONLESS MODULI

$\mathrm{Gr} = \frac{d^3 \rho^2 g \beta \triangle t}{\mu^2}$ — Grashof's Modulus, the free convection modulus.

$\mathrm{Pr} = \frac{\mu c_p g}{3600\, k}$ — Prandtl's Modulus, the fluid property modulus.

St = Stanton's Modulus, another fluid property modulus.
Pr = 1/St.

$\mathrm{Re} = \frac{a \omega d \rho}{\sqrt{2}\, \mu}$ — Reynolds' Modulus, the forced flow modulus.

No. 1130

HEATING LOW-COST HOMES

By ROBERT K. THULMAN,* WASHINGTON, D. C. AND L. E. SEELEY,** NEW HAVEN, CONN. (*MEMBERS*)

IN a paper, The Scientific Approach to the Housing Problem, by Vannevar Bush, formerly Dean of Engineering at Massachusetts Institute of Technology and now president of Carnegie Institute, Dr. Bush referring to housing says, "In our enormous country with its wide range of climate, methods of construction, and habits of the population, no single striking solution is evident or should reasonably be expected." While this statement is particularly pertinent to the heating problem, it should not mean that solutions to the problem can be made without regard to certain fundamentals. There are features upon which a substantial agreement should be reached. These general questions, if resolved, can serve as guideposts which should lead to commendable contributions in the heating of low-cost houses.

The potential market for low-cost houses warrants thought and attention in all its aspects. In the field of heating especially the opportunity exists for both constructive development and for profitable business. It is a field in which the home owner is least protected by adequate specifications, architectural service and by engineering advice. It is a market in which the operative builder avails himself of every opportunity to save money, frequently at the expense of quality.

By low-cost homes is meant those homes intended to be occupied by families with an income of less than $2000 a year. These homes may be owned or rented. They may cost $1000 or $5000. First cost is not the criterion, since success will ultimately be judged by the ability of the income group for which the housing was designed to continue to occupy the houses. The simple fact is that out of yearly income these families must be able to afford, among other things, the cost of shelter, including the cost of heat.

The following general questions are suggested as a means of formulating a common viewpoint to the heating problem:

(*a*) What standards of comfort should be secured?
(*b*) Should it be an invariable practice to provide for some minimum quantity of domestic hot water per day?
(*c*) To what extent is the building construction definitely related to heating?
(*d*) What is the proper basis for determining standards for the cost of heating?

* Mechanical Engineer, Federal Housing Administration.
** Assistant Professor of Mechanical Engineering, Yale School of Engineering, Yale University.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Mackinac Island, Mich., July, 1939.

The following answers are suggested:

First, people live in permanent and generally substantial buildings. Clothing and diet are substantially uniform. The response of human beings to their thermal environment is predictable. The AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS has established standards of comfort for human beings in enclosed spaces. Assuming that these standards are conducive to both comfort and health, it seems logical to agree that low-cost houses, like others, shall be heated in accordance with accepted standards of temperature, uniformity, air motion. humidity, etc. It follows that heating systems in low-cost houses should be capable of maintaining these standards during the heating season and that their capacity should meet the design load as determined by advocated methods of heat loss calculations. It is recognized that heating systems in many existing homes fail to secure these standards of comfort, but that is not a good argument against the standard. As a matter of fact it gives the heating industry a significant opportunity for service.

Second, it is recognized that public health is improved by increased use of water services. It is a service which does not depend upon climate or geography. The cost of hot water is more nearly a uniform cost per family than that of heating. It is suggested that a low-cost house should have from 30 to 50 gal of domestic hot water available per day. While domestic hot water need not be supplied by the heating system, there will be instances when the heating system will supply this water. In these cases the performance of the combined heating and domestic hot water plant must be considered.

Third, our climate varies so much in different sections of the country that large differences in heating costs seem inevitable. The number of degree days may vary from about 1,000 to 10,000. The variation in climate, therefore, influences the maximum capacity of a heating system over a wide range, varying about 400 per cent for a given structure. This range can be reduced by insulation, control of window sizes and generally better construction to about 100 per cent. An acceptable low-cost house can be built with maximum heat losses varying from 25,000 and 50,000 Btu per hour.

Fourth, it is an accepted fact that heating is the largest single operating cost in a house. Where small incomes must buy many necessities it is especially important that costs not only be as low as possible but that they be reasonably predictable. It is urged that heating systems for low-cost homes should not be analyzed in terms of first or initial cost alone, but in terms of operating and maintenance costs as well.

To meet the situation realistically heating costs should be expressed in terms of yearly income. If the operating cost of a heating system cannot be afforded, that system should not be installed. Operating and maintenance cost is stressed since this amount over the life of the heating equipment will be several times greater than the first cost of the system. It is suggested that the yearly budget

TABLE 1—BUDGET AMOUNTS FOR HEAT AND 30 GAL OF HOT WATER PER DAY FOR VARIOUS ANNUAL INCOMES

DEGREE-DAY	$1000	$1250	$1500	$1750	$2000
1,000	48.00	55.63	63.50	71.63	80.00
2,000	50.00	58.13	66.50	75.13	84.00
3,000	52.00	60.63	69.50	78.63	88.00
4,000	54.00	63.13	72.50	82.13	92.00
5,000	56.00	65.63	75.50	85.63	96.00
6,000	58.00	68.13	78.50	89.13	100.00
7,000	60.00	70.63	81.50	92.63	104.00
8,000	62.00	73.13	84.50	96.13	108.00
9,000	64.00	75.63	87.50	99.63	112.00
10,000	66.00	78.13	90.50	103.13	116.00

TABLE 2—BUDGET AMOUNTS FOR HEAT AND 50 GAL OF HOT WATER PER DAY FOR VARIOUS ANNUAL INCOMES

DEGREE-DAY	$1000	$1250	$1500	$1750	$2000
1,000	56.00	63.63	71.50	79.63	88.00
2,000	58.00	66.13	74.50	83.13	92.00
3,000	60.00	68.63	77.50	86.63	96.00
4,000	62.00	71.13	80.50	90.13	100.00
5,000	64.00	73.63	83.50	93.63	104.00
6,000	66.00	76.13	86.50	97.13	108.00
7,000	68.00	78.63	89.50	100.63	112.00
8,000	70.00	81.13	92.50	104.13	116.00
9,000	72.00	83.63	95.50	107.63	120.00
10,000	76.00	86.13	98.50	111.13	124.00

for heating and domestic hot water should not exceed the value given in Tables 1 and 2.

A yearly operating budget requires advance knowledge of the performance of heating devices. This means that products must be tested in a manner that will reveal pertinent economic data. A number of test codes have been devised that satisfactorily direct what measurements shall be made and in what manner. The difficulty is that these codes are not always followed. Again test data are not always published. Without test data, the economic performance of heating devices leaves too much to sheer conjecture.

The methods of determining the capacity of devices are not entirely satisfactory. Rating codes are available in some cases, but often the only indication of capacity is a catalog rating which may or may not be founded upon a generally approved basis. This suggests the desirability of a rating code for all classes of equipment and particularly a code to indicate values pertinent to actual behavior and economy of the heating system as installed and used.

Some such codes are now being prepared or revised by various groups. The practical danger is lack of uniformity of progress of the several groups or lack of uniform quality of the codes. This means that a group which presents its product in a way better suited to a determination of its economic value than some other group may enjoy a competitive advantage not entirely justified by the product itself.

The Federal Housing Administration is authorized by the National Housing Act. Its purposes are to encourage building construction and improvements in housing standards. The Act has stimulated construction of houses in the low-cost range through attractive financing terms. As a result better than 50 per cent of newly constructed houses costing under $5000 have been financed with FHA insured mortgages. Because of the large number of houses which have been submitted for mortgage insurance, it is believed that the experiences of the FHA should be of assistance in solving heating problems of the small and low-cost house field.

Before commitment to insure a property with respect to which the mortgage is issued, the FHA is required to determine the property's economic soundness.

The heating system, a vital part of the property, must, therefore, meet this test. In addition the building and its heating system must conform with standards, prescribed by the FHA, which consider health and comfort.

The amount of the mortgage which will be insured is determined by weighing various factors which determine the valuation of a property as distinguished from its cost. This system includes rating of the *mortgage pattern*, the borrower, the location, and the property itself.

Rating of the property includes rating of the mechanical equipment. The rating of this feature reflects its degree of adequacy, durability, and operating economy. This is not done by maintaining a list of approved equipment, devices, or systems. Consequently, the heating equipment or system is considered for each house on its own merits. This procedure results in the submission of a great variety of combinations of heating equipment. This necessitates more than casual knowledge of the many heating devices. In many instances catalog information is not enough and must be augmented by more specific information to enable proper consideration of a particular kind of equipment.

The establishment of standards by the FHA for other features of the house, such as space, window area, beam sizes, and plumbing, has been facilitated by reason of the existence of various codes, specifications, and other requirements. Because such codes lend themselves to the objectives of FHA, they can be incorporated directly in the requirements of the FHA with satisfactory results.

Due to lack of heating codes, however little if any protection is afforded against the use of inadequate and poorly designed heating systems, the operating cost of which may be far in excess of the low-cost house owner's ability to pay. Considerations relating to safety are covered by direct reference to the Requirements of the National Board of Fire Underwriters. These requirements are based on minimum standards for safety from fire and explosion. They are not concerned with economy of operation except insofar as safe equipment, because of inherent durability, may contribute toward lower maintenance and depreciation costs.

Concerned with a large amount of building construction, FHA receives many complaints. A major proportion of these complaints in certain localities relate to heating. Classification of the complaints on heating in one territory showed them to be due largely to excessive operating cost. These could have been avoided if technical data had been available when the commitments to insure the mortgages on these houses were made. To correct this condition, the FHA has resorted to the promulgation of specific rulings or has requested the industry whose products appear to be most directly responsible to prepare performance specifications or to revise existing codes and specifications. To avoid the liability involved in accepting inadequate or uneconomic heating systems and when the industry concerned is slow to prepare necessary corrective specifications, the FHA can avail itself of its prerogative to write down the valuation and properties. The amount of the insurable mortgage thus can be modified up or down depending upon the quality and performance characteristics of the heating system.

For example, if equipment is wisely chosen and installed, it may enhance the value of the property to the full extent of its cost. If, however, the cost of the equipment is too great in relation to the cost of the structure, or if the typical buyer cannot afford the cost of operation of the equipment, it does not add proportionately to the value of the property. These conditions must necessarily be reflected in the commitment for mortgage insurance. Obviously, the use of the ruling or request to the industry for protective standards, is more expedient and usually more effective in achieving the objectives of adequacy, durability, and operating economy.

Recently one of the Federal Housing Administration's offices received a large number of complaints on heating of small houses. It was found that in every instance an oil burner was used. The oil burner industry was requested to prepare a specification to correct the trouble. As a result of industry meetings, a program was developed which included the establishment of an installation specification and certificate of compliance. This was incorporated in a ruling requiring each installation to be tested and the results noted on the certificate.

The industry continued its program covering standards for the manufacture of the burner itself. Through the coordinating facilities of the National Bureau of Standards, the idea has been incorporated in a Commercial Standard. This procedure resulted in installation and performance codes which definitely assure improved operating economy.

A great variety of heating systems have appeared in the plans and specifications of low-cost houses. Practically every known system of heating has been utilized and there is reason to believe that each of these has its place. They include everything from the simple circulating heater to the automatic central heating plant, and include all kinds of fuels and fuel burning devices. There has been a certain geographical significance with respect to the type of heating system used, but this seems to be gradually disappearing. As a result there has been a tendency to use equipment found acceptable in a warm climate, in localities where there may be some doubt of its adequacy. This is due to the tendency to seek a cheaper system without due regard for operating economy or adequacy of performance. This trend as it develops may introduce difficulties due to performance and makes codes or standards increasingly important.

There is a trend toward the development of individual units which can be installed regardless of whether or not there is a basement. One feature of this development is the tendency to confine the equipment in a small space. Since more space is available vertically in a small house than horizontally, the procedure is to build the component parts of the unit one on top of the other. While there has been a tendency toward the elimination of the basement, this has been retarded by the complications of storage space including storage of fuel; also it is affected by construction requirements in colder climates which specify that footings shall be carried down below frost line; because of the requirements of the basementless house that the heating medium be circulated mechanically with accessories which add to the operating cost.

There are certain types of heating systems which have been characteristic of a particular locality. For example, the warm air system seems to be preva-

lent in the territory bordering on the Great Lakes and extending out into the Middle West. This may be explained by the wide public acceptance of that form of heating. This does not mean that other forms of heating which may be suitable for that territory could not be introduced. Hot water heating systems found in the houses of certain cities along the Eastern Seaboard are largely used because of public acceptance, but there is no reason why other forms of heating could not be successfully used.

Another example of the influence of public acceptance is found through the belt extending from North Carolina westward to Arkansas, where almost all houses are provided with a fireplace. As a result, the improved circulating type of fireplace finds a ready market. It has also been noted that where automatic equipment has been installed in homes already equipped with a fireplace, there has been a tendency to use the fireplace to a much lesser extent.

While there are many homes in the extreme southern parts of the country which are not ordinarily equipped with any separate heating device, there is no state of the forty-eight which has not recorded at some time or another a temperature of zero degrees or lower, and there is only one weather station in the whole United States which has never recorded a temperature below freezing. This indicates that while there may not be actual suffering in these southern homes, there is at least discomfort at times. While many of these homes do not require complete heating systems they should have some form of heating device. In Southern California it has been common practice to build houses of an *open* type of construction, eliminating the building paper and permitting a comparatively large amount of infiltration. Consequently, in Southern California, the unvented gas heater has been used to a large extent. With the use of insulating materials to provide greater summer comfort and generally tighter construction, the unvented heater presents a serious problem due to condensation. Such heaters should be separately vented or other means of heating introduced.

There is a definite trend toward the decrease of the design loss of houses by tighter construction, use of insulating materials, and more compact plans. These directly affect the maximum capacity of heating systems and particularly automatic firing devices. As a result of this trend, there is a lack of small capacity devices available, resulting in an increase in the use of small gas-fired equipment where low capacities are not a serious problem. If a continuity of heat output is to be obtained as preferred to highly intermittent heat, it would seem that the attention of the industry should be directed toward the development of devices with a capacity much smaller than those in present use.

While there is a tendency toward the development of automatic devices, this does not mean that development of devices to burn solid fuel manually should not continue. Automatic control is associated with higher cost and is limited in its application to a relatively small percentage of the possible occupants of low-cost houses. The availability of wood, particularly in rural localities, suggests the desirability of developing wood-burning equipment with definite qualities of convenience and economy. It is axiomatic that lower costs create a greater market.

Developments for heating the low-cost house will probably result in a continued improvement in all types of heating systems burning all types of

fuels. These improvements will be in the direction of reduced size, better performance, and greater operating economy. There is reason to believe that considerations growing out of reduced space and the need for privacy may result in an increased use of the combination of a heating unit with a positive distribution system so that isolated rooms may be properly heated.

A reduction in heat losses without a corresponding reduction in hot water demand has developed the use of domestic hot water as the heating medium. Although not new, this development will probably continue. While there will be further attempts to reduce heat loss, it should be pointed out that too great a reduction in infiltration loss may be made at the expense of ventilation. The question might ultimately arise as to whether or not special provisions for ventilation may not have to be made.

CONCLUSIONS

1. Agreement on certain general questions seems necessary to the basic approach to the problem of heating low-cost homes.
2. Field experience of the Federal Housing Administration reveals that the heating system is a serious concern.
3. Pertinent performance data and standards are essential to solution of problem.
4. All recognized methods of heating and their associated devices participate in the heating of low-cost houses.
5. Future development of certain devices will require that they be made for heating loads smaller than at present.

DISCUSSION

C. E. MAYETTE (WRITTEN): Individual home heating is a very definite field of unit manufacturing application, particularly from the standpoint of recurring capacities and base designs, varying mostly only as to fuel and the method of firing. Mass house heating for long term investment also taps the field of manufacturing application, though much more dependent upon developed coordinated engineering. Both applications have done a great deal to improve manufacturing, sales, distribution, and engineering practices on mechanical equipment.

H. C. MEYER, JR. (WRITTEN): The writer feels that the authors have made a real contribution to the literature on heating for low-cost homes.

The writer is connected with an engineering firm in private practice in New York City that has designed heating equipment for several of the low-cost housing projects financed with government funds, including the Williamsburg and Queensbridge projects in New York City and is, therefore, much interested in the questions discussed.

The authors apparently approve heating buildings in accordance with the Code of the ASHVE GUIDE and the writer is inclined to question whether or not a slightly lower standard would not be advisable in view of the decreased first cost and the very infrequent periods in which the lowest outside temperatures occur, based upon the theory that the lower the cost of housing the more houses can be constructed and more people provided for. In New York City, for instance, where it is customary to heat to 70 F in zero weather, the writer questions whether the installation for low-cost housing should not be designed to heat buildings to 60 F because of the fact that the temperature on so few days in any one year goes below 10 F.

While the authors were apparently discussing the heating of small houses, it has been the writers' experience that in large housing projects, especially in buildings

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of four stories and over in height, the vacuum heating system with some simple form of controlling the heat from a central point in each building, will produce a sufficient reduction in operating cost to pay the fixed charges on the added cost of such a system over the cost of the cheapest kind of one-pipe steam job.

The writers' office recently roughed out a design for a heating plant for a group of three- and four-story buildings containing a total of 456 apartments. Consideration was given to a central steam plant with controlled heat, the use of three plants with controlled heat, and a one-pipe job with boiler in each building, fuel oil being considered in each case. The results of the study showed a decided preference in favor of the central plant, not only in initial but in operating cost as well. On the other hand, it is the writer's belief that very large central heating plants are not apt to pay because of the great cost of the distributing system for carrying steam in considerable quantities to points remote from the boiler house.

Williamsburg, comprising 1614 apartments is provided with 10 boiler plants for a total of 20 buildings with runs between buildings of comparatively short length. Queensbridge, with 3161 apartments and a total of 26 buildings located on six city blocks, is provided with six boiler plants, each block being provided with its own plant.

The writer wishes to conclude with the statement that while a good deal has been accomplished in the matter of heating low-cost housing, a good deal has yet to be done, as the authors point out in this very excellent paper.

J. C. FITTS (WRITTEN): The ASHVE has taken a very important first step in the solution of the problems surrounding the heating of low-cost homes.

I hope that the activities of the Society in this field will not be limited to listening to this paper, applauding it and then dropping the subject. It would seem that this topic would furnish an excellent theme for chapter meetings where the problem could be discussed by architects, engineers, manufacturers and contractors with the thought of working out local problems in connection with heating in this field as well as in contributing to the general discussion. The authors of the paper are to be complimented on taking as one of the principles to govern such heating installations that the operating cost is of far more importance than the first cost.

It has been my belief for many years that those who install heating systems in houses, especially in small homes, should so design the system as to be sure that it was adequate and would operate economically not alone as to fuel cost but also as to repairs. By proper installation domestic hot water can be made available in these homes from the heating plant with a minimum of operating expense.

The authors have pointed out the importance of codes and standards. In this connection the attention of the members is called to the program of certified heating which has been developed by the *Heating, Piping and Air Conditioning Contractors National Association,* in which the home owner is assured by the local association that the system installed is adequate in accordance with well established standards.

The encouragement of the membership of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS would be helpful in spreading the certified heating program and making this service available to home owners throughout the United States.

W. E. STARK (WRITTEN): The authors have made an excellent presentation of a subject that is of vital interest to the heating industry. Residential heating is the industry's bread and butter today, and indications are that low-cost homes are going to be the predominating component of residential building for some time to come.

One who builds his own home covered by FHA mortgage insurance or who purchases such a house from a speculative builder, is assured of reasonable protection against low quality, quick depreciation and high maintenance in practically every

component of the structure with the exception of heating. As the authors point out, adequate codes exist governing such things as plumbing, fire protection and various structural members and these are incorporated in FHA requirements. However, nothing of a basic and widely recognized character exists with respect to heating, so that the house owner has little assurance either of immediate satisfaction or of protection against recurrent maintenance and replacement costs which will increase his cost of ownership beyond the limits of his budget. The use of automatic and mechanical equipment is penetrating down into these low-cost homes and the purchasers of such homes are being done an injustice when they are not protected on their heating equipment to the same degree that they are protected on such things as plumbing, relation of floor space to occupants, window area, beam sizes, etc., etc. The home owner can be sure that his floor will not collapse under the weight of a piano or sure that enough sunlight will come in to keep the children from getting the rickets, but he cannot be sure that his heating plant will not fall down completely in the midst of winter, due possibly to faulty rating, selection, or installation of equipment; or to breakdown of flimsy fuel handling, fuel burning, or air circulating equipment; with resultant detriment to the health of the family.

In spite of the trend toward mechanical heating in homes of all cost categories, it is questionable whether there should not be a line drawn somewhere that will protect the owner of a low-cost home in the maintenance charges which even the best of mechanical and automatic equipment is occasionally subject to.

Various trade and manufacturers' associations have made substantial contributions to the protection of the home builder. However, each of them operates in a limited field. The ASHVE, on the other hand, is interested in the entire field, but only from a broad humanitarian and social standpoint. Commercially, the Society is totally disinterested and is, therefore, qualified to take a leading part in the rectification of the unsatisfactory condition the authors so ably point out. Although the Society has done much good in the instigation and promulgation of standards governing such things as comfort air conditioning, still such standards do not have the same broad social value that standards of home heating would possess.

In the totalitarian nations the home and family have ceased to be the fundamental units of society. In our land they remain fundamental and the perpetuation of this condition is being encouraged every time a low-cost home is built which will make it possible for a family to have a little piece of ground with a comfortable building on it which they can call their home. Our organization can make an equal contribution by taking the steps that are necessary to insure that these homes are truly comfortable and healthful places in which to live.

T. H. URDAHL (WRITTEN): It is through such papers as this that a more tangible interpretation of the value of the Society's work for general public good is clearly brought before the member. Here for the first time the Government strives to protect as well as foster home ownership through the promulgation of standards in sharing safety and economic stability, not so much with a paternalistic attitude, but on the basis of its being sound business judgment to protect the investment for which the Government and synonymously the tax payer is the guarantor, through making available building funds on a long time repayment basis. We are fortunate in having Mr. Thulman and Professor Seeley as members whose cooperative effort in this paper is so helpful in giving direction to necessary elements of research, standardization, and improvement in heating facilities.

PROF. C. M. HUMPHREYS: The heating system for a low-cost home should be adequate to properly heat the home, but should not be oversized to the extent that the initial cost is unnecessarily high. It is only by careful design that the heating system for a small home can be made economical in both first cost and operating cost.

Today the heating system for the average small home is designed and installed by a contractor who knows little or nothing of modern design methods. Sometimes these contractors are unscrupulous fellows who install undersized systems made of inferior materials. In this case the home owner may have saved in first cost but will pay dearly in operating and maintenance costs and in dissatisfaction.

Many of the contractors doing small home heating are honest and conscientious, but lack the knowledge necessary to design economically. These men usually install oversized systems which will add unnecessarily to the first cost of the home.

The average man contemplating the construction of a home will find that a small amount of money spent for engineering services will prove to be a good investment. It is almost sure to save him money, either in first cost or operating cost.

C. TASKER: The suggestion has been made that in estimating the heating load for low-cost homes a saving in first cost can be made by departing from the accepted standards and calculating the load by taking the outdoor temperature as say 10 deg above the figure usually used. This it is claimed will still give satisfactory heating and result in a substantial saving in first cost and possibly in operating cost. I would suggest that this is very likely to lead to a considerable amount of dissatisfaction on the part of the owner.

The heat losses from low-cost houses due to transmission through the walls and ceiling are likely to be low at first since in most cases savings in first cost and operating cost will be sought by installing insulation wherever economically possible. These losses may not increase unduly with the life of the building. Heat losses due to infiltration however will not in all probability be high at first and will certainly not be calculated on the high side, but they are likely to increase considerably as the age of the building increases. In the low income groups little, if any, money is available for maintaining the structure in the best condition and it is more than likely that the deterioration which will undoubtedly take place will show itself first in increased infiltration losses. If the heating plant has been *skimped* because the calculated heat losses were too small; unsatisfactory heating will soon result. If the heating plant installed is fully adequate to take care of correctly calculated heat losses with an ample margin of safety, it is much more likely to give *consumer satisfaction* for a reasonable period of years.

C. F. MALLY: It appears that while there is a great need for heating units that are directly applicable to the low-cost homes, it is important also that quality, good design and proper installation should become an integral part of this discussion. Due to the extreme competition and desire for low cost, furnaces are installed that are not even good stoves and are made of material not suitable. A heating unit should last at least 10 years or perhaps as long as the mortgage.

While the government sets down definite requirements as to guarantee, this guarantee is too general to become effective. The work of the *Oil Burner Institute*, the *American Gas Association, National Warm Air Association* and the ASHVE is excellent and should eliminate many evils. It seems a difficulty arises in enforcing these recommendations as set down by these bodies. Perhaps cooperation with local community codes and the FHA district offices, with competent engineers who are in the field, should bring about conditions which will establish simple and clear cut standards. At present local community codes only eliminate fire hazards and should be made to consider also the quality, effectiveness and satisfaction of the installation from the standpoint of the tax payer.

No. 1131

EVALUATING PERFORMANCE OF WATER HEATERS FIRED WITH SOLID FUELS

By H. J. Rose,* (*MEMBER*) and R. C. Johnson,** Pittsburgh, Pa.

More than 200,000 coal and wood water heaters for domestic hot-water supply are produced annually in this country. This is more than the number of house-heating boilers that are produced,[1] so that methods for evaluating the performance of such equipment should be of interest to all who design, select, install and operate such equipment.

Water heaters are commonly rated and sold in terms of the output of hot water produced at their full burning rate. The householder is also interested in such factors as the amount of attention required, uniformity of water temperature, length of time that the fire will bank without dying out, and economy. However, there is a lack of published methods for testing solid-fuel-fired water heaters so that they can be compared in terms of consumer satisfaction.

Several years ago the writers were engaged in the engineering development of water heaters having improved performance characteristics. In order to evaluate the development models, and to compare them with existing types, it was necessary to devise test methods. These measured not only the output at continuous full burning rate, but also flexibility of output and response to automatic controls, banking characteristics, amount of attention required, performance of heater and storage tank in meeting simulated household demands, and other factors.

After the heaters were tested by these methods in the engineering laboratory, they were placed in homes, and instrument records were kept of their performance over a considerable period of time. In this way it was found that the tests described in this paper give a reliable index of the performance that can be expected in actual household use.

* Senior Industrial Fellow, Anthracite Fellowship, Mellon Institute of Industrial Research.
** Industrial Fellow, Anthracite Fellowship, Mellon Institute of Industrial Research, and former Research Chemist, the Philadelphia and Reading Coal and Iron Co., Pottsville, Pa.

[1] Census of Manufacturers: 1937, Heating and Cooking Apparatus, Except Electric. (*U. S. Bureau of the Census, Industry Release No. 1119*, Dec. 6, 1938.)

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Mackinac Island, Mich., July, 1939.

Rate of Hot-Water Output

All water heaters, regardless of the kind of fuel used, may be classified into three general types with regard to flexibility of hot-water output as follows:

a. Instantaneous heaters—Require no storage tank as the water is heated as fast as it is drawn.

b. Quick-recovery heaters—Used in connection with storage tanks, the size of which depends upon the flexibility of hot-water production and the maximum quantity of water required during periods of demand.

c. Slow-recovery heaters—Have a low maximum hourly output and require either a large storage tank in which to accumulate a liberal reserve supply of hot water, or else a low demand rate.

There is no sharp dividing line between quick-recovery and slow-recovery water heaters, and the methods given can be used for testing both types.

This paper does not attempt to set up performance specifications or standards since this is a function of appropriate standardizing groups. Instead, it presents a correlated series of tests which give rather complete information on the performance of water heaters, both by themselves and in connection with storage tanks. The information obtained is particularly valuable for giving designers and manufacturers an accurate idea of heater performance. For some purposes, only a part of the tests need be made.

Apparatus Required

Fig. 1 is a schematic arrangement of the set-up used. All of the tests can be made with this equipment, although not all of it is needed for each test. The heater is shown on platform scales with flexible water connections and a sliding stack connection. This permits the fuel consumption to be determined at frequent intervals. An alternate method eliminates the scales and flexible connections, in which case the fuel consumption during a test is determined by shaking the grates and filling to the original level with a weighed amount of the fuel.

Stack Drafts

A definite stack draft is maintained in all of the methods described, although in actual household use, heaters are subject to variable natural draft conditions, which are influenced as much by the chimney and by weather conditions as by the characteristics of the heater. Since natural chimney draft fluctuates too much to be very suitable for comparative engineering tests, it has been found preferable to apply a definite draft to the heater.

It was found that a draft of 0.03 in. water is suitable for practical testing, but tests have been made at drafts of 0.01, 0.05 and sometimes 0.07 in. of water.

Maximum Output

This test measures the hourly rate of hot-water production from a running start, when the fire is burned at a definite applied draft. It does not differ materially from some methods that are in use for determining the hot-water

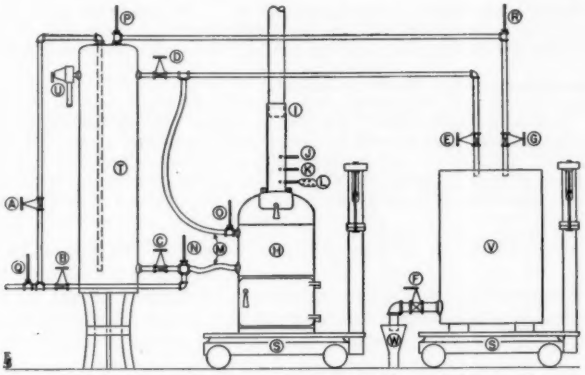

FIG. 1. SET-UP FOR TESTING SOLID-FUEL-FIRED WATER HEATERS

Key to Fig. 1

A, B. Valves on cold-water inlets to storage tank and heater.
C, D. Gate valves of ¾ in. or larger size, depending on heater output.
E. Needle valve with graduated wheel for close adjustment of flow during output tests.
F. Quick-opening gate valve 1½ in. or larger, for emptying weighing tank.
G. Gate valve used for draw-off tests.
H. Hot-water heater.
I. Stack with slip joint to permit weighing heater.
J. Gas-sampling tube.
K. Draft tube.
L. Thermocouple or thermometer for determining stack temperature.
M. High-pressure flexible connections to permit heater to be weighed.
N, O, P. Thermometers, preferably of recording type.
Q, R. Thermometers.
S. Scales for weighing heater and weighing tank.
T. Hot-water storage tank of selected size.
U. Temperature and pressure relief valve piped to drain.
V. Weighing tank.
W. Drain.

Note: Draft dampers and thermostatic control are not shown in the drawing, since these vary widely in type and location.

output for rating purposes. Only the general procedure will be outlined, since it is not the purpose of this paper to discuss the details of testing technique.

1. By-pass the hot-water storage tank and provide a slow continuous flow of water through the heater and the weighing tank to the drain. This is done by closing valve *A*, opening *B*, closing *C* and *D*, opening *E* slightly, and opening *F*.

2. Start a fire in the heater with paper and charcoal or other kindling, and fill with the solid fuel which is to be used in the test.

3. Apply a fixed draft of 0.03 in. water, measured in the stack close to the breeching.

4. Adjust valve *E* as often as necessary to maintain a temperature rise of 100 F in the water passing through the heater. When the heater appears to have reached its full output, shake the grates until red coals appear, remove ashes, fill the heater with fuel, record the weights of heater and weighing tank *V*, close valve *F* and record the time of starting the test.

5. Continue to maintain a temperature rise of 100 F through the heater by adjusting valve *E*, and weigh the water discharged at 15- or 30-min intervals. When the weighing tank is full, empty it by means of the large quick-opening valve *F*, noting the time required for drainage, so that a correction can be made for the inflow during the time that valve *F* is open. (An alternate method would be to provide two weighing tanks.)

6. If the heater is on scales *S*, as shown in Fig. 1, the weight should be recorded at hourly intervals to determine rate of coal consumption.
7. Obtain stack temperatures throughout the test, at *L*, preferably with a recording instrument.
8. It is desirable to obtain stack gas analyses.
9. Continue the test until the heater needs refueling as determined by the condition of the fire. If it is on scales, record the loss in weight. Shake the grates until red coals appear, remove the ash, and fill the heater with fresh fuel to the original level, and reweigh. (If the heater is not on scales, the amount of fuel burned is determined from the weight of fuel required to refill.) The carbon content of the ash should be determined. The heater is now ready for the pick-up test procedure given in par. 10.

The output per hour in gallons of water of 100 F rise should be plotted against time. A secondary scale showing the total Btu introduced into the water per hour may be provided on the graph. Efficiency should be reported as the percentage of the fuel's heat content which appears in the hot water. The gallons of 100 degree-rise water produced per pound of fuel may also be reported.

The test shows the output of the heater hour by hour, over the entire period from fueling to refueling. Commercial ratings of water heaters vary with respect to the temperature rise used and the length of time over which the output is averaged. For engineering comparisons, complete data are desirable.

Pick-up of Fire Which Has Required Refueling

The maximum output test described was made with a running start, whereas this test measures the rate of pick-up of a fire which has needed refueling and has had the maximum amount of fresh fuel added. This simulates a common household condition.

10. Start with the freshly tended fire described in par. 9, and continue the use of 0.03 in. water draft and 100 deg rise, weighing the water at 15-min intervals until full output is reached.

Rate of Decrease of Output When Fire Is Checked

As soon as the heater reaches full output in the pick-up test, the fire is checked, and the rate of decrease in output is determined. This measures any tendency for an overrun in storage-water temperatures, and the manner in which the heater will respond to control.

11. Starting with the heater producing its full output, as obtained in par. 10, damper off the fire to the fully banked position without changing the stack draft, and determine the output by the same procedure at 15-min intervals until a fairly constant low output is reached.

Minimum Output

As soon as the low banking rate has been reached, the heater is left in this banked condition over night. If desired, the output obtained during this period may be determined, but this is not always done since the actual outputs at the beginning and end of banking are regularly obtained.

12. As soon as a low banking rate has been obtained according to par. 11, the heater is left in a banked condition for 12 hours without attention and without

changing the rate of water flow. If the output is not determined during this period, it should be determined for one hour at the end of the test.

Pick-up After Banking

This test measures the responsiveness of the heater when the dampers are opened after a long banking period without giving any attention to the fire.

13. Continuing from par. 12, set the dampers in the full burning position but do not shake the grates or add coal. Continue the use of 0.03 in. water draft, maintain a 100 deg temperature rise, and determine the output at 15-min intervals, until a satisfactory output is obtained, or until more fuel is needed (as determined by the condition of the fire).

Graph the results of the tests covered by pars. 10 to 13 inclusive on a single sheet, plotting gallons of 100 degree-rise water against time. A supplementary Btu scale may be included on the graph. The results of the foregoing tests will give an accurate idea of the maximum and minimum outputs of the heater, and its response to damper control at the stated draft with the particular fuel used.

Length of Time That Heater Can Be Banked

It is desirable for water heaters to be able to remain banked for several days without danger of the fire dying out. This permits the family to be absent over the weekend, for example, and to have a tankful of hot water and the fire in good condition upon returning. The water should not overheat during banking periods, for several important reasons.

Inexpensive thermostatic damper controls are available for solid-fuel-fired water heaters. If the heater does not come equipped with such a control, one should be installed for the remaining tests.

In this test, the heater is connected with the minimum recommended size of storage tank, and the fire banked under thermostatic control, without drawing off any water, and without giving any attention to the heater until more fuel is needed. The purpose of this test is two-fold. It measures the maximum length of time which the fire can be banked without attention, and shows whether there is any tendency for the water to overheat in a minimum size tank, when no water is drawn off.

The authors have tested small water heaters burning anthracite fuel, which could be banked continuously for four days or more without danger of the fire dying out, and without any tendency to overheat the water in a 30 gal, fully insulated tank. Larger heaters of similar design have been banked for 10 days or more with the same results. At the end of these long banking periods, the fire was still in good condition. This indicates what can be expected from solid-fuel-fired water heaters which have been carefully designed on the basis of reliable test data.

14. As soon as the test described in par. 13 is completed, shake the grates, remove the ash, fill the heater with fuel, weigh, and connect it to the storage tank which is filled with cold water. This storage tank should be of the minimum size which is recommended for use with the particular heater, and should be provided with the

maximum insulation (if any) which is recommended. The change-over to the tank is made by opening valve *A*, closing *B*, opening *C* and *D*, and closing *E*.

An appropriate thermostatic damper control should be used. This control keeps the damper in the burning position until the tank is filled with hot water, after which it will function from time to time to keep the water temperature up to the desired level. Temperatures at the top of the hot-water storage tank should be recorded throughout, preferably by a recording thermometer.

No hot water should be drawn off and no attention whatever given to the fire until the end of the test, which occurs when more fuel is needed, as determined by the condition of the fire.

15. Determine the weight of fuel consumed. Shake the grates until red coals appear, remove ash, and fill with fuel.

Report the length of time that the heater can be banked under these conditions; the amount of fuel consumed; and graph the temperature at the top of the hot-water-storage tank throughout the test.

Tests Simulating Household Conditions

All water heaters (except those of the instantaneous type) are used with storage tanks. Since satisfactory performance under typical household conditions depends to an important degree upon the size and location of the tank, the damper controls, etc., as well as upon the water heater itself, it is essential to determine the actual performance of a particular combination that is recommended for general use.

Hot-water demands in private homes vary greatly from hour to hour, being characterized by heavy draw-offs at certain times, and periods of many hours when there is little or no demand. This increases the difficulty of designing hot-water heaters as compared with the design of hand-fired house-heating boilers where the changes in load are more gradual.

For simulated household draw-off and banking tests, the heater is connected with a tank of recommended size, and a thermostatic damper control is used. Widely varying amounts of hot water are drawn off hour by hour, according to definite schedules based on hot-water demands in typical homes. The draw-off schedule is continued without any attention being given to the fire until the hot-water system fails to meet the scheduled demand.

Average hot-water demands per family range from less than 50 gal to more than 200 gal per day, depending upon the size of family, ages and personal habits of the members, number of hot-water outlets, number of servants, standard of living, etc. However, even in the case of families with low average consumption, there are periods of peak demand. Storage tanks of less than 40-gal capacity do not provide enough hot water at one time for certain household uses, regardless of the type of heater or the fuel used (whether solid, liquid or gaseous).

Nevertheless, such a large proportion of the tanks sold are of the 30-gal size that heaters have been regularly tested connected to tanks of this capacity as well as larger sizes. Any heater which gives acceptable results when connected with a 30-gal tank will give much better performance with a larger

tank, with regard to availability of hot water, avoidance of overruns in water temperature during banking, etc.

The combined draw-off and banking tests are made as follows:

16. The test is started from conditions existing at the end of par. 15, after the fire has been tended and as soon as the water in the tank is at the desired storage temperature. The thermostatic damper control is left in operation, and a stack draft of 0.03 in. water maintained.

17. Hot water is drawn off through valve *G* in accordance with the 50-gal schedule given in Table 1. Since the water will not always have a 100 deg rise in temperature, the actual quantity to be drawn off must be calculated. The most convenient method is to draw off a weight of water at the observed temperature which will contain the desired number of Btu. In Table 1 it is assumed that within the temperature range

TABLE 1—DRAW-OFF SCHEDULE FOR 50 GAL DEMAND[a]

HOUR OF DRAW-OFF	WATER GAL OF 100 F RISE	BTU IN DRAW-OFF
8 a.m.	15	12,255
9 a.m.	6	4,902
10 a.m.	3	2,451
11 a.m.	1	817
12 m.	3	2,451
1 p.m.	9	7,353
2 p.m.	3	2,451
3 p.m.	2	1,634
5 p.m.	8	6,536
	50	40,850

Repeat the schedule on the following day.

[a] *Note:* This draw-off schedule has been condensed to nine hours, whereas hot-water demands in the typical home spread over about 16 hours. This condensed schedule has been used because it fits in well with the regular testing day, and it has been found practical in comparing and evaluating hot-water systems. This schedule is of course more severe than drawing off the same quantity of water over a period of 16 hours, and it thus represents more nearly the rate of demand on peak days.

used, each gallon of water contains 8.17 Btu per degree temperature rise. The most convenient method is to draw off a weight of water at the observed temperature which contains the specified number of Btu. The temperature of the water discharged is observed continuously during the draw-off, by thermometer *R*. If the temperature changes much, the water is weighed in increments so that the heat content can be accurately calculated. The temperature of incoming cold water *Q* must also be recorded during this test.

18. The 50-gal draw-off schedule given illustrates how it applied to a particular laboratory working day.

19. Larger draw-offs may be made, especially with heaters or tanks of higher capacities. In this case proportional amounts are withdrawn; for example, a 100-gal schedule is obtained by doubling the quantity of water drawn off each hour in the 50-gal schedule.

20. At the end of each day's schedule draw-off, the heater is left under thermostatic control until the following morning, when the same schedule is repeated. This cycle is continued until the system is unable to meet the full scheduled demand with hot water of usable temperature (*e.g.*, 100 F or more). No attention is given to the fire at any time after starting the test. Determine the amount of fuel consumed.

In reporting these tests, the following should be given: The number of days during which the 50-gal or 100-gal (or larger) schedule was met without any attention to the heater, the total gallons of hot water of specified temperature rise which were drawn off during the test, and the pounds of fuel consumed. It is convenient to report efficiency as gallons of 100 degree-rise water produced per pound of fuel burned, during the entire test. The combined draw-off and banking test, as described before, forms an important practical method for evaluating the over-all performance of a water heater and storage tank. It is often desirable to make this test first, so that a good idea of characteristic performance and limitations can be obtained, before the more specific tests are authorized.

SUMMARY

This paper briefly describes methods by which solid-fuel-fired water heaters (for domestic hot-water supply) may be tested for maximum and minimum output, flexibility of output or response to control, maximum attention intervals and efficiency. A combined draw-off and banking test is described which simulates household demands, and gives a good idea of the over-all performance which can be expected in actual use.

Field installations of water heaters based on knowledge gained by these tests have proved so satisfactory, in comparison with previous methods of water heating used in the same homes, that these test methods should be of engineering interest.

DISCUSSION

A. J. JOHNSON (WRITTEN): The increased interest which our Society is taking in the problems of the solid fuel industry suggests the possibility that some solution or action can be taken to unravel one of our most perplexing problems in connection with water heating. I refer to the fact that it is almost universal practice to rate a line of tank heaters (commonly called bucket-a-day-stoves) in direct proportion to their height and weight. For example, a line of tank heaters 12 in. in diameter will be rated in such a manner that the smallest, or namely the one with the shallowest fuel bed, will have the lowest rating and the largest or the one with the deepest fuel bed will have the highest rating.

With the diameters the same, the only difference between these two extreme heaters from a combustion angle is obviously the fuel bed depth. Thus, from the angle of combustion rate and consequent output, the smallest heater in reality will always show greater outputs than the larger ones of the same diameter. For this reason, tests such as Mr. Rose has described never substantiate the manufacturer's rated capacities of such heaters. Yet, in spite of the fact that such tests are obviously correct, at the same time no one manufacturer is in the position to completely revise his system of rating unless this is done simultaneously throughout the industry. The obvious result is a stalemate with manufacturers unwilling to have tested ratings published and with a continuation of a policy of rating heaters on a basis of their cast-iron weights rather than upon any standard of actual performance.

This fact is so pronounced that it is preventing the entrance of any engineering into the rating of tank heaters and thus would seem to be worthy of the serious consideration of the Society.

E. T. SELIG, JR. (WRITTEN): In the past, overemphasis has been placed on the rating, or output of domestic water heaters, and very little attention paid to such items of consumer satisfaction as convenience, ability to bank for long periods without overheating or the fire dying out, flexibility, frequency of attention, dependability, etc.

This paper fully appreciates that domestic hot-water demand in a residence is extremely variable, and that the ability of a water heater to meet a steady demand hour after hour is only one of a number of essential features of good water-heater performance. I believe that if water heaters were designed with a better appreciation of performance requirements, as brought out by the test methods suggested in this paper, better service would be provided to the residential consumer.

As pointed out by Colonel Danielson in his paper at the Society's Annual Meeting in Pittsburgh, January, 1939, it is the purpose of the Research Technical Advisory Committee on Solid Fuels to study the utilization of solid fuels with the ultimate objective of providing better service to the residential and small commercial consumer. It is the hope of this committee that papers dealing with the performance of other types of solid-fuel-burning equipment, such as the one prepared by Dr. Rose and Mr. Johnson, may be presented at future meetings of the Society.

W. A. DANIELSON (WRITTEN): Simulating service conditions is a difficult matter and the authors are to be congratulated on the results they have obtained. The *American Gas Association* found it necessary to test gas water heaters under similar rules. Prior to this change, the apparent efficiency was high in that it was obtained at maximum rating without including stand-by losses. The actual service efficiency was nearer 30 per cent generally than the more than 70 per cent claimed. If testers generally would follow the rule set forth in this paper much better equipment would result. The manufacturers should be compelled to place a name plate on the equipment that would give the essential information as is the case with gas equipment.

The maintenance of a draft of 0.03 in. during pick-up is not exactly simulating service conditions. The pick-up will be much slower, as it will be necessary to heat up the chimney to obtain adequate draft. If the lining of the chimney can be kept separate from the outside masonry covering, then the pick-up will be more rapid. This phase is being studied by the Committee on Solid Fuels in the research work now under way at Primos, Pennsylvania.

Most testing of solid fuel equipment is done with anthracite. There the greatest portion of heat release is radiant. Greater combustion volume is, of course, required with bituminous coal, and some provision should be made in the testing rules to provide for the use of this kind of fuel. The writer made a number of installations a year ago with heaters that had many times the combustion volume of the ordinary heater and the results were very much better than was expected.

JOHN JAMES (WRITTEN): Accurate methods of rating the output and performance of small domestic units from an engineering viewpoint are urgently needed in the solid fuel industry. It is hoped that the presentation of such papers as this one will stimulate the thinking of the Society membership to encourage the development of standards and codes whereby the residential consumer will have some measure to compare the performance of small domestic solid fuel-fired equipment.

The cooperation of divergent viewpoints in development of adequate standards frequently results in the improvement of products by the entire manufacturing indus-

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try. This progress results from accurate determinations of output and subsequent improvements in design to develop higher efficiencies.

Th procedure as outlined by the authors in this paper seems to include all of the essential features in ascertaining the performance of water heaters fired with solid fuels.

No. 1132

OZONE IN VENTILATION—ITS POSSIBILITIES AND LIMITATIONS

By WILLIAM N. WITHERIDGE,* DETROIT, MICH., AND CONSTANTIN P. YAGLOU,** (*MEMBER*), BOSTON, MASS.

THE purpose of this work was to study the possibilities and limitations of ozone for counteracting body odors in ordinary occupied rooms. Answers were sought to the following questions:

(a) How much ozone is required to counteract body odors under various practical conditions?

(b) How are the occupants affected by such ozone concentrations?

(c) Does ozone oxidize body odors in the air, or does it "mask" them under practical conditions of ventilation?

The action of ozone on various odors, such as hydrogen sulfide, butyric and valeric acids, indol, skatol, ammonia, stale urine, and others, has been studied by several investigators, but to our knowledge, there are no quantitative data on the effect of low ozone concentrations upon body odors in the air of ordinary occupied rooms. A serious obstacle to these studies has been the lack of a sensitive chemical method for measuring such low ozone concentrations as are permissible in ventilation work. Moreover, knowledge is still lacking on the constituents that make up the *body-odor complex*, and the only basis to work on at present is the smell of the complex and the quantity of clean air required to dilute it to the threshold concentration.

Low ozone concentrations in rooms estimated from the ozonator output and extent of dilution are open to criticism because of the great variability in decomposition of ozone, depending on the characteristics of a room and of the air supply system, absolute humidity which appears to be very important, temperature, and possibly other factors that may arise under practical conditions. Body odors, too, have peculiar disappearance characteristics of their own, and it is difficult to predict their interaction with ozone except by actual tests using the human nose for judging the results.

In the present study, low ozone concentrations were estimated from the

* Industrial Hygiene Engineer, Detroit, Mich.
** Associate Professor of Industrial Hygiene, Harvard School of Public Health.

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strength of smell, using a technic developed in previous works.[1, 2, 3] An interesting finding which permitted estimation of ozone concentration in the presence of body odors was the inability of the latter to affect the ozone smell. With proper technic, there was no difficulty in distinguishing ozone intensities regardless of body-odor strength.

REVIEW OF PHYSICAL, PHYSIOLOGICAL AND PHARMACOLOGICAL PROPERTIES

Ozone consists of three atoms of oxygen in an unstable combination, represented as $O_3 \rightarrow O_2 + O$, and the oxygen atom set free possesses great oxidizing

TABLE 1—EFFECTS OF OZONE IN VARIOUS CONCENTRATIONS

PARTS OZONE PER MILLION PARTS OF AIR (BY VOL.)	OBSERVED EFFECTS	AUTHORITY
0.01	Threshold for keen sense of smell	Hill's Discussion (19) (Confirmed by present study)
0.015	Normal threshold; readily detectable by most normal persons	Present study
0.04	Upper allowable limit in occupied rooms	Present study
0.10	Objectionable to all normal persons; irritating to mucous membranes of nose and throat in most persons	Present study
0.5–1.0	Disorders breathing, reduces oxygen consumption and shortens lives of guinea pigs	(5), (8), (13)
0.3–1.5	Inhibits fungus and mold growth in cold-storage rooms	(14), (15)
1–10	Headache, respiratory irritation, and possible coma	(5), (6)
15–20	Lethal in 2 hours to small animals	(5)
>1,700	Lethal in a few minutes	(8)
6,500	Germicidal for air-borne organisms	(15)

power. Ozone is a normal constituent of country and polar air in detectable, but ill-defined, concentrations. It is produced in nature photochemically by ultra-violet light, and is almost absent in city air, except during thunderstorms. Even in the absence of oxidizable material, the rate of decomposition of ozone is known to increase appreciably with a rise of humidity and to a smaller extent with temperature rise.

Ozone has a pungent odor, somewhat similar to that of chlorine or garlic, and is, therefore, a very effective *masking* agent for odors which are not readily oxidized. In high concentrations it readily oxidizes organic matter and has a variety of applications in water sterilization, bleaching, and in the control of fungi and spores in cold-storage rooms. The oxidation of body odors by ozone at concentrations that are permissible in ventilation work is highly debatable. Erlandsen and Schwarz,[4] Hill and Flack,[5] Jordan and Carlson,[6] Konrich,[7] and others working on odors that are supposedly related to the body-

[1] Reference Numbers refer to Bibliography.

odor complex, ascribed the counteraction to *masking,* rather than to oxidation. These studies were severely criticized by the ozone industry, directly or indirectly, but no authoritative data have been presented to refute the findings.

The toxicologic properties of ozone are fairly well known. Even in relatively low concentrations (0.1 to 1 ppm) it exerts a powerful irritating action on the mucous membranes of the respiratory organs and shortens the lives of guinea pigs after months of continuous exposures.[8] Pneumonia is the terminal condition. Much higher concentrations lead to death in shorter periods from lung congestion and edema.[5 to 9] There is no systemic poisoning, as most of the ozone is decomposed on the mucous membranes of the respiratory tract, where its destructive action is mainly localized.

The irritation threshold appears to be about 0.05 ppm; there is no evidence of damage to health in concentrations well below this. Workers in the London subways have shown no ill effects after years of exposure to concentrations at or slightly above the olfactory threshold. In Table 1 are outlined the effects at various concentrations, as reported by different investigators.

It is now generally recognized that ozone is not a satisfactory germicide at concentrations within the range of human tolerance (0.01 to 0.04 ppm); its effectiveness varies with the concentration, the degree of atmospheric humidity, and last, but not least, with the species of bacteria involved.[5, 6, 9, 10, 11]

On certain species, Jordan and Carlson[6] were unable to obtain any sure germicidal action with amounts of ozone from 3 to 4.6 ppm. Concentrations sufficient to kill dry typhus bacilli, staphylococci, or streptococci in the course of several hours killed guinea pigs sooner in Sawyer's experiments.[9]

Claims for converting carbon monoxide to carbon dioxide by the use of ozone in garages are now generally discredited.[12]

METHOD OF PRESENT STUDY

Description of Experimental Rooms: The experiments were carried out in the spring and summer of 1936. Two adjacent rooms were used, each having a net air space of about 1,400 cu ft. One of the rooms designated hereafter as the *experimental* room, was occupied by the subjects who produced the odor to be counteracted by ozone. The other room served as a *control* room for pure air and was occupied by not more than two *judges* at a time who estimated the odor intensities in the experimental room. A small tight door, closing against felt cushions, allowed passage of the judges from the control to the experimental room for estimating the ozone or body-odor intensity and quality of air. All window and door cracks in the experimental room were sealed with plasticene and taped over with adhesive tape. In tests with low airflows, the small connecting door was kept closed in order to avoid inward leakage and the observations of odor intensity were made in the exhaust outlet outside the experimental room. With high airflow, door leakage was outward and did not affect the results.

Both rooms were ventilated by downward supply systems, and the quantity of air introduced was measured by thin-plate orifice meters. The used air

escaped to the corridor through sensitive check louvers near the floor of the rooms. Air supply to the experimental room was varied in different experiments, according to the number of occupants and the desired body-odor intensity. In the control room, the air supply was always high enough to keep the body-odor intensity at or below the olfactory threshold. Separate air conditioners installed in the corridor kept the two rooms at a comfortable temperature and humidity.

Ozone Production and Measurement: Ozone was generated inside the experimental room by corona discharge using an electric precipitator similar to that of Drinker's for the quantitative determination of atmospheric dusts.[16] The air to be ozonized was filtered and heated always to about 90 F, as shown in

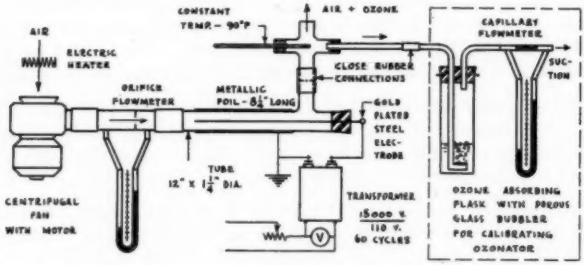

FIG. 1. APPARATUS FOR PRODUCING AND MEASURING OZONE

Fig. 1. All parts of the apparatus were kept scrupulously clean to preclude production of nitrogen oxides.

The output of the ozonator was measured by oxidation of potassium iodide and titration of the liberated iodine with sodium thiosulfate using starch as an indicator.[17] The arrangement of apparatus for producing and measuring the ozone is shown diagrammatically in Fig. 1. Ozone output was controlled by varying the transformer primary voltage, the airflow through the ozonator, and the length of the inner electrode. Fifty-five different sets of conditions yielded the following empirical equation from which the ozonator output could be predicted within the experimental range:

$$\text{Cfm } O_3 = 53\ (V - 75)\ (250 - G)\ (Q + 25)\ (L + 8) \times 10^{-13}$$

where:

V = transformer primary voltage (80 to 110)
G = absolute humidity, grains per pound dry air (4 to 120)
Q = airflow through ozonator, cubic feet per minute (2 to 10)
L = length of precipitator inner electrode, inches (4 to 8)

The ozonator was located in the back of the experimental room behind the subjects, and the ozonized air was blown upward against the incoming air so as to obtain good mixing.

RESULTS

Olfactory Ozone Intensity in Relation to Ozone Concentration: The success of the whole study depended largely on the ability of the human nose to judge low ozone concentrations that are below the limits of present chemical methods of analysis. It was necessary, therefore, to study first of all the relationship between olfactory intensity and concentration of ozone in clean air free from body odor.

Ozonized air in measured quantities and concentrations was diluted to known extents with the air supplied to the room, the mixing taking place inside the experimental room. After allowing sufficient time for equilibrium, two trained

TABLE 2—OLFACTORY SCALE OF ODOR INTENSITIES

ODOR INTENSITY INDEX	DESIGNATION	EXPLANATION OR QUALIFICATION
0	None	No perceptible odor
½	Threshold for keen sense of smell	Very faint, barely detectable by trained persons, imperceptible to untrained persons
1	Normal threshold	Detectable by all normal persons but not objectionable
2	Moderate	Neither pleasant nor disagreeable; allowable limit in rooms
3	Strong	Objectionable; air regarded with disfavor
4	Very strong	Disagreeable, annoying, irritating
5	Overpowering	Sickening, nauseating

judges passed from the control to the experimental room several times in each test and recorded the strength of ozone, immediately after entering, according to the scale in Table 2. The results are shown in Fig. 2, and it will be seen that for any given ozonator output, the ozone concentration, and hence its olfactory intensity in the room, depended largely on the absolute humidity which affects the ozone decomposition rate. In a series of tests with absolute humidities of 3 to 5 gr (grains) per pound of air, obtained by silica-gel dehydration, ozone was found to be fairly stable after equilibrium was reached and the concentration in the air was but slightly lower than that computed from the ozonator output. This agrees with previous work on ozone decomposition in *bone-dry* air free from oxidizable material.[18]

Approximate ozone concentrations corresponding to any given ozone intensity in the experimental room can, therefore, be read off the 0 humidity line in Fig. 2. The amount of ozone to be added to the air in the room in order to obtain the given ozone intensity, or concentration, depends on the absolute humidity and can be read off the appropriate humidity line in Fig. 2. For example, to obtain an odor intensity of 2, the required ozone concentration must be about 0.03 ppm. If the humidity is 40 gr per pound of dry air the amount of ozone to be added to the air to obtain an equilibrium concentration of 0.03 ppm with a corresponding olfactory intensity of 2, is 0.10 ppm, whereas

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with a humidity of 100 gr per pound the ozone to be added must be approximately 0.9 ppm. These data apply only to the experimental room used in this study under conditions of equilibrium.

Neutralization of Body Odors by Ozone: The method of carrying out tests was somewhat similar to that followed in previous work on ventilation requirements,[2, 3] except for the addition of ozone near the middle of each test. Experiments ran from 9 a.m. to 12:30 p.m. or 1 to 5 p.m. Altogether, 95 persons served as subjects, largely adults and grade-school children chosen from the poorest socio-economic status.

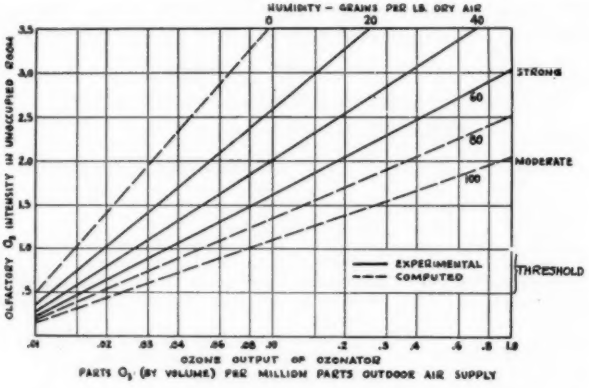

FIG. 2. EFFECT OF HUMIDITY ON DISAPPEARANCE OF OZONE IN UNOCCUPIED EXPERIMENTAL ROOM

The lines for 0, 80, and 100 grains humidity were extrapolated from the data for the 4, 20, 40, and 60-grain lines using the following empirical equation:

$$R = 3.08 - \frac{\log (13.3\ G + 250)}{1.155}$$

n which G represents the humidity in grains per pound of dry air and R represents the ratio between ozone intensity at humidity G and ozone intensity at zero humidity on the same ordinate. The average deviation of the experimental data from the values given by this equation was 0.07 on the odor scale, and the range of variations was from 0.12 positive to 0.23 negative on that scale.

After the body-odor intensity was raised to equilibrium during the first part of each test, the ozonator was turned on and adjusted so as to get maximum body-odor obliteration with the minimum smell of ozone. Ozone and body-odor intensities were estimated by two to six *judges* upon entering the experimental room from the control room, using the olfactory scale in Table 2 for both odors.

Fig. 3 shows the progress of a typical experiment. The ozonator was turned on about an hour after the subjects entered, and in 20 min the body-odor intensity dropped from 3.5 to 0.5, while the ozone intensity kept on rising toward the objectionable point. When the ozone output was reduced, the body-odor intensity rose simultaneously, and after about an hour, both approached equilibrium.

A summary of the data obtained in fourteen tests under equilibrium conditions is given in Table 3. It can be seen in columns 6 and 7 that the introduction of ozone reduced the body-odor intensity in most instances, but in no case was it possible to counteract the two odors so that neither could be smelled. Ozone concentration had to be perceptible to do any good, and the higher the body-odor intensity, the higher the ozone intensity had to be to counteract it. The limiting factor was the ozone odor itself which sometimes became as objectionable as the body odor.

It would seem, therefore, that the use of ozone alone is impractical for counteracting strong body odors and that it must be supplemented by the

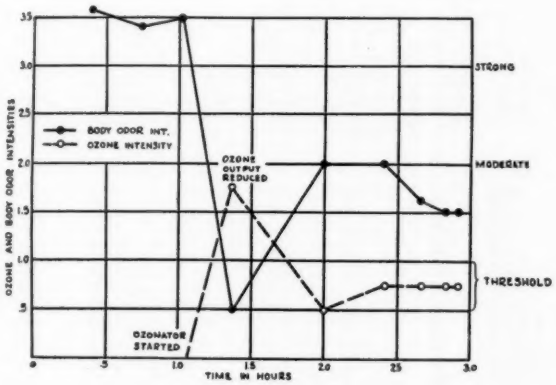

FIG. 3. RECORD OF AN EXPERIMENT SHOWING EFFECT OF OZONE ODOR ON BODY-ODOR INTENSITY. ROOM AIR SUPPLY AND NUMBER OF OCCUPANTS CONSTANT

introduction of sufficient outside air to lower the body-odor intensity to 3.5 or less. Small quantities of ozone can then be used to advantage for reducing the odor from 3.5 to 2 or less. Under the experimental conditions, an ozone odor intensity of 0.75-1.00, (0.015 ppm approximately), reduced the outdoor air requirement for the control of body odor more than 50 per cent, as can be seen in Fig. 4. Ozone intensities less than 0.75-1.00 had no effect on body odors, while higher intensities proved to be unpleasant to a few sensitive individuals.

Masking vs. Oxidation: The results of this study would seem to indicate that the action of ozone on body odors takes place not in the air of the room by oxidation, as is sometimes assumed, but on the mucous membranes of the nasal passages where the ozone is decomposed. If the reduction were due to oxidation in the air, some of the ozone would have been utilized in the process and the ozone intensity would have been lower with subjects than without subjects in the room, when all other conditions remained unaltered. No significant loss

of ozone appears to have taken place, as can be seen in columns 5, 6 and 7 of Table 4. Positive signs in column 7 indicate loss, and negative signs indicate gains of ozone, the latter due, of course, to errors. The mean loss in all 14 experiments is practically zero, in fact negative, indicating no detectable chemical action.

If the odorous emanations of the human body consist largely of acids, as is generally believed, little oxidation can be expected from the small quantities of ozone used in the experiments. On the other hand, the olfactory threshold concentration of ozone is low enough, by comparison with that of most body

TABLE 3—COUNTERACTION OF BODY ODORS WITH OZONE
(*Summary of Experimental Data Under Conditions of Equilibrium*)

1 Expt. No.	2 No. of Subjects	3 Air Temp. Deg F	4 Humidity Grains H_2O/lb Dry Air	5 Outdoor Air Supply cfm/ Person	6 Initial Body Odor Intensity (no O_3)	7 Final Body Odor Intensity with O_3	8 Observed Olfactory Ozone Intensity	9 Ozonator Output ppm. by Vol. Outdoor Air Supply	10 Estimated Ozone Concentration in Room Air ppm. by Vol. (from Fig. 2)
82	7	78.0	33	17	2.0	0	1.75	0.044	0.023
83	7	75.4	30	13	3.0	0	2.25	0.087	0.036
84	7	82.0	77	21	1.5	1.5	0.50	0.028	0.012
85	7	85.0	98	7	3.0	2.25	0.75	0.039	0.012
86	9	82.7	70	10	3.0	1.75	0.75	0.026	0.012
87	7	85.5	87	13	3.25	2.0	1.00	0.024	0.011
88	7	78.8	48	6	3.5	2.0	1.00	0.054	0.020
89	7	81.5	65	6	3.5	0.75	1.50	0.083	0.020
90	8	83.5	90	15	3.5	1.5	0.75	0.074	0.016
91	15	81.0	78	<5	4.25	0.5	2.50	0.412	0.034
92	7	82.0	88	6	4.0	1.5	1.75	0.183	0.021
93	7	81.2	58	15	3.25	3.5	0.50	0.021	0.012
94	7	81.5	80	16	2.5	2.5	0.50	0.028	0.012
95	7	77.0	74	22	2.5	1.0	0.88	0.016	0.010

odors, to readily mask the latter, with the possible exception of the odors of skatol and indol.

Even hydrogen sulfide, which appears to be the most readily oxidizable substance in the known group of body odors, can only be masked in ventilation work because the concentration of ozone required for complete oxidation is prohibitive from the standpoint of both toxicity and smell. The normal olfactory threshold (odor intensity 1) of hydrogen sulfide is placed at 0.002 mg per liter of air. An objectionable odor strength of 3 would, therefore, correspond to $0.002 \times 2^2 = 0.008$ * mg per liter approximately. To oxidize this hydrogen sulfide would theoretically require $0.008 \times \frac{48}{34} = 0.0113$ mg of ozone per liter of air, or $0.0113 \times 509 = 5.8$ ppm. Actually the amount of

* Concentration corresponding to an odor intensity index n = olfactory threshold concentration $\times 2^{n-1}$

† 48 = molecular weight of ozone.

‡ 34 = molecular weight of hydrogen sulfide.

§ 1 mg of ozone per liter of air at 70 F = 509 ppm by volume.

ozone required for complete oxidation must be considerably in excess of this already prohibitive concentration. On the other hand, about 0.1 ppm of ozone (0.0002 mg per liter) appears to be sufficient to mask a hydrogen sulfide odor intensity of 3. This quantity of ozone could theoretically oxidize a maximum of $\frac{34}{48} \times 0.0002 = 0.00014$ mg of hydrogen sulfide or only $\frac{0.00014}{0.0113} \times 100 = 1.2$ per cent of the hydrogen sulfide in the air.

Using even much greater ozone concentration in an experimental chamber, Erlandsen and Schwarz [4] failed to obtain any demonstrable oxidation of

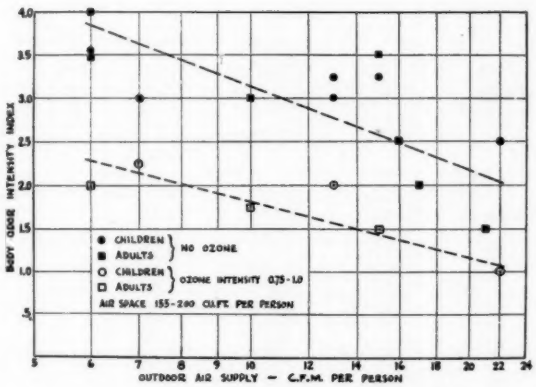

FIG. 4. EFFECT OF A MILD OZONE ODOR ON BODY-ODOR INTENSITY AND ON AIR REQUIREMENTS FOR BODY-ODOR CONTROL

hydrogen sulfide, ammonia, trimethylamine, butyric and valeric acids, indol and skatol, although the odors of these substances were effectively masked.

Further evidence in support of the *masking* effect is the reappearance of body odors with disappearing ozone odor at the end of an experiment (see Fig. 5) after the subjects had left the room—the ozonator and air supply having been turned off and the room closed tight. Referring to Fig. 5, the ozone intensity just before the subjects left was high enough to obliterate completely the body odor, but when the ozone intensity fell to the threshold, the body odor reappeared and continued to increase, gradually approaching the normal body-odor disappearance curve (dotted line) without ozone. The implication here is that ozone was incapable of destroying body odors in the air while the subjects were in the room, despite its high concentration. Odoriferous material was adsorbed on wall surfaces and when the subjects left, desorption began and continued as its usual normal rate, as if no ozone had been added.

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POSSIBILITIES AND LIMITATIONS

From the standpoint of ventilation, it should make no great difference whether ozone masks or actually destroys body odors in the air. If it improves the quality of air and does no harm to the occupants, it should be of some value, and under favorable conditions it has proved to be of value in this and other previous studies.[19]

Although the *judges* were extremely sensitive to the odor of ozone upon entering the experimental room (containing ozone) from the control room

TABLE 4—LOSS OF OZONE-ODOR INTENSITY IN COUNTERACTING BODY ODORS

1 EXPT. NO.	2 ROOM ABSOL. HUMIDITY GRAINS/LB AIR	3 OZONATOR OUTPUT PPM. BY VOL. OUTSIDE AIR SUPPLY	4 ESTIMATED OZONE CONCENTRATION IN ROOM AIR PPM. BY VOL. (FROM FIG. 2)	5 OBSERVED OZONE INTENSITY WITH SUBJECTS IN ROOM (FROM COL. 8, TABLE 3)	6 CORRESPONDING OZONE INTENSITY WITH NO SUBJECTS IN ROOM (FROM FIG. 2)	7 LOSS OF OZONE INTENSITY (COL. 6-COL. 5)
82	33	0.044	0.023	1.75	1.60	−0.15
83	30	0.087	0.036	2.25	2.20	−0.05
84	77	0.028	0.012	0.50	0.70	+0.20
85	98	0.039	0.012	0.75	0.70	−0.05
86	70	0.026	0.012	0.75	0.70	−0.05
87	87	0.024	0.011	1.00	0.60	−0.40
88	48	0.054	0.020	1.00	1.40	+0.40
89	65	0.083	0.020	1.50	1.40	−0.10
90	90	0.074	0.016	0.75	1.10	+0.35
91	78	0.412	0.034	2.50	2.10	−0.40
92	88	0.183	0.021	1.75	1.50	−0.25
93	58	0.021	0.012	0.50	0.70	+0.20
94	80	0.028	0.012	0.50	0.70	+0.20
95	74	0.016	0.010	0.88	0.40	−0.48
Mean				1.17	1.13	−0.04

(containing no ozone), most of the subjects in the experimental room were unaware of ozone intensities of 1.0 or less corresponding to ozone concentrations of 0.015 ppm or less, except when the concentration was built up too rapidly. Many subjects noticed an improvement in air freshness at such low ozone concentrations, while a few complained of a *strange odor in the air.*

Ozone intensities over 2 (0.04 ppm or more) were associated with dryness of the upper respiratory mucosa, and distinct irritation at higher concentrations. It appears, however, from Table 3 and Fig. 4, that it is rarely necessary to use ozone intensities greater than 1.0 (0.015 ppm). This concentration can barely be smelled by the occupants of a room, yet it permits a considerable reduction in the fresh-air supply (see Fig. 4) and there is no evidence that this amount is harmful even on prolonged exposures, as for instance in the London subways.

The main objection to the application of ozone to ventilation is lack of

control. Although the output of commercial ozonators may be controlled adequately by dehydrating and filtering the treated air, no means are available for regulating the actual room concentration which may vary within wide limits according to humidity (see Fig. 2) and general characteristics of a room. A given ozone output at the machine may yield only a trace of ozone in a room on a humid day, but the same output in dry weather may result in a most undesirable condition. The danger is especially great in central station systems that have sufficient capacity to make up for loss of ozone in transit to the rooms.

Under such conditions, ozone may do more harm than good and its use should be discouraged, at least until adequate means are devised for accurately con-

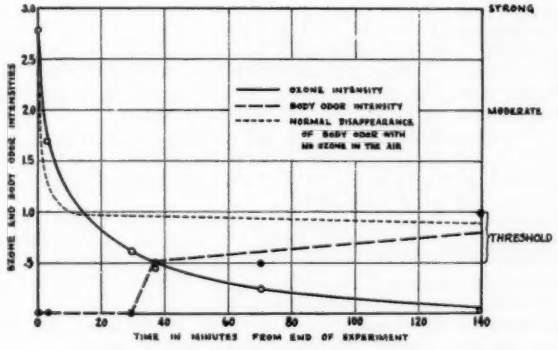

FIG. 5. REAPPEARANCE OF BODY ODOR WITH DISAPPEARING OZONE ODOR IN TIGHTLY-SHUT ROOM AT THE END OF AN EXPERIMENT AFTER SUBJECTS LEFT ROOM

trolling the concentration in occupied rooms. The so-called *ionizing* machines which have recently appeared produce less ozone than the old ozonators and their effectiveness for *masking* odors is correspondingly smaller. The ions produced by these machines have little effect on odors and *freshness* of air. The underlying agent is ozone and the problem of concentration control still remains unsolved.

SUMMARY

Ozone in concentrations of 0.015 ppm, that barely can be smelled by the occupants of a room, reduced the smell of body odor sufficiently to permit a reduction of at least 50 per cent in the fresh-air requirement for odor control. Higher concentrations were irritating to the mucous membranes of the upper respiratory tract, while lower concentrations had no effect on body odors. The action of ozone on body odors appears to take place not in the air of the room by oxidation, as is sometimes assumed, but on the mucous membranes of the nasal passages by *masking*. Although ozone in sufficient strength completely obliterated the smell of body odors, it proved difficult to counteract the two

so that neither could be smelled by persons entering the room from fresh air. Ozone had to be present in perceptible concentrations to do any good.

The main objection to the application of ozone to ventilation is shown to be lack of control. Until adequate means are devised for controlling the output of ozonators from the actual concentration in occupied spaces, the use of ozone should be discouraged because of its great toxicity.

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17. Analytical Chemistry, by Treadwell and Hall. (Vol. 2, Quantitative, 6th ed., John Wiley & Sons, New York, 1924.)

18. Der Thermische Ozonzerfall, by Hans-Joachim Schumacher and G. Sprenger. (*Zeitschr. f. Physikal Chemie,* Abt. B, Vol 6, 1929-1930, p. 446.)

19. Ozone in the St. Louis Schools, by E. S. Hallett. (ASHVE TRANSACTIONS, Vol. 28, 1922, p. 33.)

DISCUSSION

DR. HENRY F. SMYTH, JR.[20] (WRITTEN): I have been in contact with research work which was planned to develop a method for determining ozone in the presence

[20] Industrial Fellow, Mellon Institute of Industrial Research, Pittsburgh, Pa.

of nitrogen dioxide, which is the form in which it occurs in practically, if not all, electrical generations of ozone. The conclusion of this very thorough study was that it was, so far, impossible to determine ozone quantitatively in the presence of nitrogen dioxide because of the difficulty, if not impossibility, of distinguishing between very small amounts of the two vapors. For this reason, I question any work in which ozone concentrations in the atmosphere are reported, because I am convinced that any figure for ozone in the atmosphere includes an unknown proportion of nitrogen dioxide.

If this feeling of mine is correct, it becomes apparent that we have no quantitative information on the effects of ozone alone. What information we have may be correct for the effects of the combined gases generated by various ozonizers, and therefore may be adequate to interpret the performance of these ozonizers if it be recognized that this performance be due to a combination of vapors.

It is, I believe, universally conceded that high concentrations of ozone are irritating. However, I believe that the irritating qualities of any ozonizer can be judged only by actual experience with that ozonizer and with a particular set of operating conditions.

I do not want to throw doubt on any work in the performance of a particular instrument, but merely to throw doubt on any quantitative arguments as to the effect of measured concentrations of ozone. I believe the question as to whether the use of ozonizers is desirable or not is not yet settled. I feel sure that if ozonizers are used, the concentration of their mixed product must be kept at or below the odor threshold. Whether their operation at this low rate materially influences foreign odors in a room, I am not prepared to say.

DR. M. B. FERDERBER (WRITTEN): In the past several years a few fallacies concerning the use of ozone as a deodorizer resulted in the manufacture and sale of equipment designed to cleanse the atmosphere of odors resulting from smoking, body odors, etc. It was not unusual to find "ozone generators" in smoking rooms and other places where persons congregate. There followed a period of discredit, apparently based on good scientific background that ozone could not actually destroy odors. Recently, however, there have been many inquiries in the medical field regarding ozone generators, which might be used to destroy the odors of necrotic and putrefying tissues found in certain severe infections and abscesses. From actual observations the ozone produced was extremely irritating to the patient and the masking effect described by the authors is certainly evident and the substitution of one irritant for another would not produce any criteria for using such devices. The complaints of patients in such an atmosphere consisted of lightheadedness, irritation of air passages, with some nausea. These complaints would not justify using such a procedure even if it were of some value. Dr. Smyth has based his information on his own and others' experiences in the research field and of ozone determinations. I feel that additional information may add a great deal to the evidence presented by the authors in a simple, concise work of its kind.

W. N. WITHERIDGE: Dr. Smyth is of course right in stating that nitrogen dioxide is produced during the electrical generation of ozone. However, as a result of tests made by the U. S. Bureau of Mines in cooperation with the ASHVE on four types of ozone generators, Yant reported in 1923 that in no case were oxides of nitrogen found in the ozonized air in proportions greater than 1 part to 25 parts of ozone. Generators of the brush discharge type were found to produce approximately 1 part oxides of nitrogen to 150 parts of ozone. Hartman reported that the ratio did not exceed 1 part oxides of nitrogen to 100 parts of ozone on generators he used.

XUM

The corona discharge type of apparatus used in our experiments (Fig. 1) had been studied earlier by Drinker and others [21] and the amounts of oxides of nitrogen it produced at room temperatures were found to depend on voltage, tube diameter, and air velocity. Under proper conditions the amounts of oxides of nitrogen produced were so small as to be undetectable by chemical methods. In the light of this earlier work the voltages and air velocities applied to the ozonator used in the present study were at all times adjusted so that nitrogen oxides would not be produced in more than negligible quantities.

As both Dr. Ferderber and Dr. Smyth point out, there are still several problems associated with the use of ozone in ventilation that have not been solved. Some of these items can be left properly to the fields of chemistry and medicine for study. Others are of sufficient practical importance to the air conditioning industry and profession to deserve further consideration by workers in this field.

PROF. C. P. YAGLOU: Dr. Smyth's skepticism of possible effects of NO_2 in our experiments can be allayed by pointing out that NO_2 is relatively harmless at low concentrations. It takes as much as 62 parts of NO_2 per million parts of air to cause irritation to the throat [22] whereas less than 0.1 parts of ozone is sufficient to irritate. The maximum allowable concentration for prolonged exposures is 39 ppm for NO_2, as against 0.04 ppm for ozone. Even if we assume that our ozonator produced as much as 1 part of NO_2 in 150 parts of ozone, the concentration of NO_2 in the air would be $\frac{0.04}{150} = 0.0003$ ppm, which is beyond the limits of human perception and of present methods of chemical analysis.

[21] Loc. Cit. Note 16.
[22] U. S. Bureau of Mines, Technical Paper No. 248.

No. 1133

EFFECT OF SIZE AND TYPE OF AIR INLET AND OUTLET ON THE HEAT OUTPUT OF CONVECTORS

By A. P. KRATZ,* M. K. FAHNESTOCK,** AND E. L. BRODERICK,*** (*MEMBERS*)
URBANA, ILL.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the University of Illinois.

THE object of this investigation was to determine the effect of different sizes and types of inlets and outlets on the heat output of convectors, and, if possible, to correlate the areas of such inlets and outlets with the free area through the heating unit. All tests were run in a warm wall test booth and in every case the distance between the back of the convector cabinet and the wall was 2½ in. For the principal series of tests three convectors were used, each consisting of a different type of heating unit fitted into a special cabinet which was similar to a commercial cabinet except that it was provided with a means for adjusting the widths of inlets and outlets. In all cases a grilled front outlet was used in connection with either a grilled or an ungrilled front inlet, and a program of tests was conducted to determine the performance of the convectors with different combinations of widths of inlets and outlets. Additional tests were run in order to determine the effect of reducing the length of the inlet relative to the length of the cabinet, of introducing different arrangements of vanes in connection with the inlets and outlets, of using an inlet at both the front and back of the cabinet, and of locating the outlet in the top of the cabinet. Some tests were also run on a special convector having a steam heated humidifying pan in the upper part of the cabinet.

DESCRIPTION OF APPARATUS

The warm wall test booth was constructed in accordance with the specifications given in the ASHVE Standard Code for Testing and Rating Concealed

* Research Professor, Engineering Experiment Station, University of Illinois.
** Research Assistant Professor of Mechanical Engineering, University of Illinois.
*** Research Assistant in Mechanical Engineering, University of Illinois.

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XUM

Gravity Type Radiation (Steam Code)[1] and has been described in previous papers.[2, 3, 4] An elevation section of the test booth and a detail of the piping are shown in Figs. 1 and 2. For most of the tests the piping used differed slightly from that shown in the Code, in that the by-pass for determining the piping correction was omitted. In some of the earlier tests this by-pass was

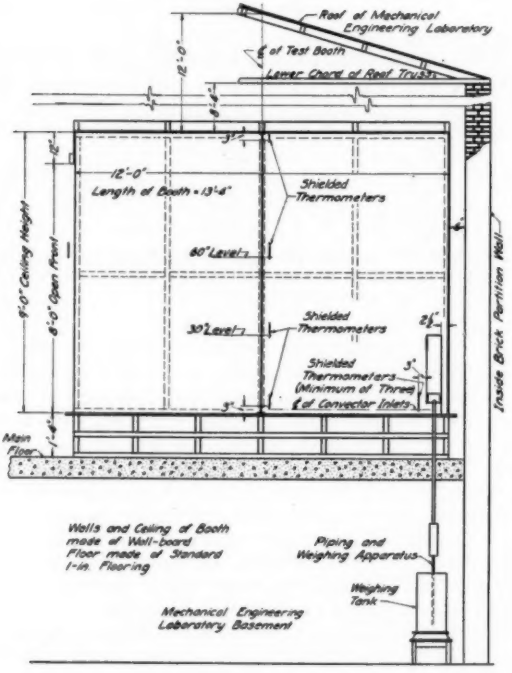


FIG. 1. SECTION OF WARM WALL TEST BOOTH

used, but it later proved to be unnecessary. For all later tests the piping correction was found to be constant, and was obtained by means of separate tests, for the purpose of which a length of heavily insulated pipe was substi-

[1] ASHVE Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code). (ASHVE TRANSACTIONS, Vol. 37, 1931, p. 367. Revised January 1936, ASHVE TRANSACTIONS, Vol. 42, 1936, p. 29.)

[2] ASHVE RESEARCH REPORT No. 938—Tests of Convectors in a Warm Wall Testing Booth, by A. P. Kratz, M. K. Fahnestock, and E. L. Broderick. (ASHVE TRANSACTIONS, Vol. 38, 1932, p. 511.)

[3] ASHVE RESEARCH REPORT No. 963—Tests of Convectors in a Warm Wall Testing Booth, Part II, by A. P. Kratz, M. K. Fahnestock, and E. L. Broderick. (ASHVE TRANSACTIONS, Vol. 39, 1933, p. 319.)

[4] ASHVE RESEARCH REPORT No. 998—Factors Affecting the Heat Output of Convectors, by A. P. Kratz, M. K. Fahnestock, and E. L. Broderick. (ASHVE TRANSACTIONS, Vol. 40, 1934, p. 443.)

tuted for the convector. As compared with the apparatus specified in the Code, a more accurate determination of the water weights was made possible by using a receiver having a section of ¾-in. pipe instead of 2-in. pipe placed opposite the section of the gage glass used for reading the condensate level. The steam pressure was more accurately obtained by using a water manometer instead of one filled with mercury.

The three types of heating units used for most of the tests are shown in Fig. 3 together with one of the special cabinets. The heating unit for Convector No. 35 consisted of finned cast-iron elements bolted together, and had a nominal height of 6 in. The heating unit for Convector No. 36 consisted of three round copper tubes and for Convector No. 37 of four oval copper tubes

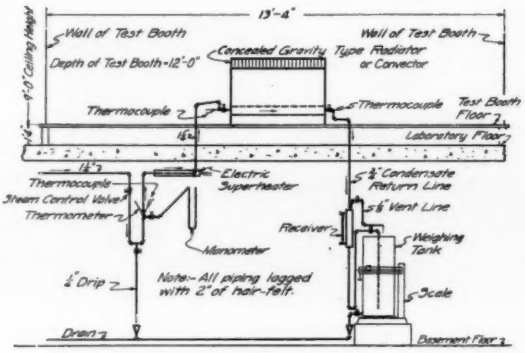


FIG. 2. DETAIL OF PIPING FOR WARM WALL TEST BOOTH

each having attached fins and extending between cast-iron headers. The height of the heating unit for Convector No. 36 was 2 in., and for Convector No. 37 was 5¼ in.

The principal dimensions of the three convectors are given in Fig. 4. The cabinets for these convectors were of the same height and, since the heating units were of approximately the same length, the lengths of the inlets and outlets were practically the same. The area of the inlet or outlet was varied by maintaining the length constant and changing the width. In the case of the inlet a plate partly covered the opening as shown in Fig. 3 and the width of the opening was increased by raising the plate. The maximum width was obtained when the lower edge of the plate was on a level with the bottom of the heating unit. A similar plate with a serrated upper edge partly covered the outlet and was lowered in order to increase the width of the opening. The type of grilles employed, as shown in Fig. 3, had ¾-in. slots and ¼-in. frets. The widths of inlets and outlets used in the test program are given in Fig. 4.

Details of the special arrangements used in connection with tests not included in the principal test series are shown in the insets appearing in conjunction

with the different curves representing the test results. (See Figs. 10, 11, and 12.)

Modifications of Convector No. 35 were used for all of the tests run with a reduced length of inlet and with different arrangements of vanes at inlets and outlets. (See Fig. 10.) Cardboard cut to the proper shape and size was used to block the ends of the inlets and sheet metal was used for the different vanes which were fastened in place with metal screws. A vane used at the top of the outlet was formed with a radius equal to the inside width of the cabinet, and one-half of the width was employed for the radii of intermediate vanes used at the center of the inlet and outlet. In one case a special vane was placed just below the outlet and was arranged to form a venturi throat.

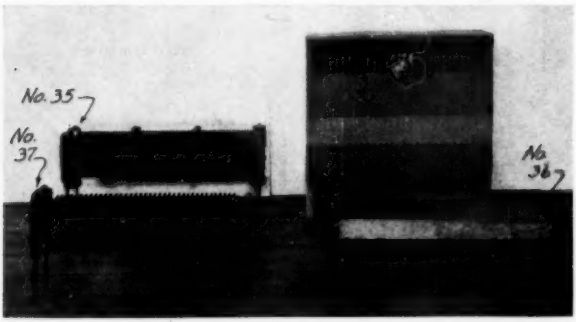


FIG. 3. CONVECTOR CABINET AND THREE TYPES OF HEATING UNITS

This vane was 1½ in. wide and reduced the width of the cabinet at that point from 5¾ in. to 4¼ in.

Two convectors were used for the tests on the arrangements having outlets in the top of the cabinets. For the tests conducted with modifications of Convector No. 35 (See Fig. 11) the front outlet was covered, and a grilled outlet similar to the front outlet was provided in the top of the cabinet. This convector had a heating unit 6 in. in height. Convector No. 33, which was used for a second series of tests on a convector having a top outlet, had a cast-iron heating unit 12 in. in height. Convector No. 31 was used for a series of tests on a cabinet having both a front and back inlet. The heating unit was made of cast-iron and was 4½ in. high. The cabinet had a conventional inlet in the front, below the heating unit, and a corresponding inlet in the back. One series of tests was run with both inlets open, and one series with the back inlet closed.

Convector No. 11*a* was a modification of Convector No. 11. The heating unit, used in connection with a commercial cabinet, was the same heating unit as that used for Convector No. 37. In Convector No. 11*a* a water pan, or humidifier, was used at the top of the cabinet, and the outlet was made correspondingly wider. This water pan had a false bottom which formed a steam

space that was connected by pipes at both ends to the steam headers of the heating unit. (See Fig. 12.) The water in the pan was thus evaporated with heat furnished by the condensation of steam in the space between bottoms. Details of this arrangement are given in the insert shown in connection with the curves representing the results.

TEST PROCEDURE

All tests were run with a steam temperature of 216.5 F in the convectors. The relation between the heat output, or the heat equivalent of the net steam condensed, and the temperature of the air at the inlet was established by running tests at different inlet air temperatures varying over a range of from 60 to 80 F. In order to obtain this range of inlet air temperatures, the large laboratory in which the test booth was located was heated or cooled so that the desired inlet air temperature resulted when equilibrium had been attained.

No test observations were made until conditions had remained constant for several hours, as indicated by readings of all thermocouples and thermometers. The condensate was then weighed over the period of one hour, and no test was accepted if the condensate showed more than 2½ per cent deviation in the successive 10-min increments of weight. The total condensate for each test was corrected by subtracting the condensation due to the piping alone as determined by separate tests.

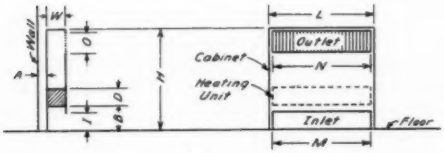
The tests on each of the three principal convectors were divided into four series as indicated in Fig. 4. For Series I the width of the grilled outlet of the cabinet was set at 5 in., and ungrilled inlets of different widths were used. The setting of the grilled outlet remained the same for Series II, but a grilled inlet was used in each case instead of an ungrilled inlet. In Series III and IV the width of the inlet remained constant and different widths of grilled outlets were used. For Series III an ungrilled inlet was used, and the width of this inlet was the same as that of the grilled inlet used in Series IV. The width of the grilled inlet selected for Series IV was the same as that of the grilled inlet which gave the best performance in Series II.

No change was made in procedure for any of the other tests with the exception of the tests on Convector No. 11*a*, which had the steam heated water pan in the upper part of the cabinet. During all of the tests on this convector the liquid level in the water pan was observed by means of a gage glass connected to the water pan, and the temperature of the water, or the temperature at which evaporation occurred, was read by means of a thermometer inserted through the top of the cabinet. The weight of water evaporated was obtained from the readings of the gage glass, which was calibrated in terms of the weight of water in the pan.

RESULTS OF TESTS

General Results

The effect of the widths of the inlets and outlets on the performance of convectors was determined by comparing the heat outputs for the various arrangements. This heat output was expressed in terms of the heat equivalent



Convector Number	Type of Heating Unit	Dimensions in Inches								Heating Surface in Sq. Ft.		
		A	W	H	L	D	B	M	N	Primary	Secondary	Ratio*
35	Cast Iron	2 1/4	5 1/4	29	34 1/2	6	7	32 1/4	31 1/16	5.9	17.4	3.3
36	Non-ferrous	2 1/4	5 1/4	29	30	2	6 3/4	27 1/4	27 1/4	1.7	22.4	13.2
37	Non-ferrous	2 1/4	5 1/4	29	29 1/4	5 1/4	7	27	26 3/8	3.5	29.2	8.3

*Ratio of Secondary to Primary Heating Surface

Convector Number	Series Number	Dimensions of Inlets and Outlets																	
		Inlet Width in Inches = I, Free Area in Sq. In. = F								Outlet Width in Inches = O, Free Area in Sq. In. = F									
33	I	Ungrilled	I	2	3	4	5	6	7	Grilled	O	5							
			F	64.5	96.8	129.0	161.3	193.5	225.8		F	116.1							
	II	Grilled	I	2	3	4*	5	6		Grilled	O	5							
			F	44.1	68.1	92.1	116.1	140.1			F	116.1							
	III	Ungrilled	I	6						Grilled	O	2	3	4	5	6	7	8	
			F	193.5							F	44.1	68.1	92.1	116.1	140.1	164.1	188.1	
	IV	Grilled	I	6						Grilled	O	2	3	4	5	6	7	8	
			F	140.1							F	44.1	68.1	92.1	116.1	140.1	164.1	188.1	
36	I	Ungrilled	I		3	4	5	6	6 3/4	Grilled	O	5							
			F		83.6	111.5	139.4	167.3	188.2		F	101.6							
	II	Grilled	I		3	4	5	5 3/4		Grilled	O	5							
			F		57.6	80.6	101.6	116.1			F	101.6							
	III	Ungrilled	I	5 3/4						Grilled	O		3	4	5	6	7		
			F	160.3							F		57.6	80.6	101.6	122.6	143.6		
	IV	Grilled	I	5 3/4						Grilled	O		3	4	5	6	7		
			F	117.4							F		57.6	80.6	101.6	122.6	143.6		
37	I	Ungrilled	I		3	4	5	6	7	Grilled	O	5							
			F		81.0	108.0	135.0	162.0	189.0		F	98.0							
	II	Grilled	I		3	4	5	6		Grilled	O	5							
			F		57.5	77.7	98.0	118.2			F	98.0							
	III	Ungrilled	I	6						Grilled	O		3	4	5	6	7		
			F	162.0							F		57.5	77.7	98.0	118.2	138.5		
	IV	Grilled	I	6						Grilled	O		3	4	5	6	7		
			F	118.2							F		57.5	77.7	98.0	118.2	138.5		

FIG. 4. DIMENSIONS OF PRINCIPAL HEATING UNITS TESTED

of the net steam condensed under standard conditions. The latter, as defined by the ASHVE Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code),[5] involves steam in the convector at 215 F and air entering the inlet at 65 F, or a standard temperature difference of 150 F. Since it is not always possible to run a test with a given arrangement under exactly standard conditions, one of two methods may be used to correct the heat output obtained under test conditions to the equivalent heat output with

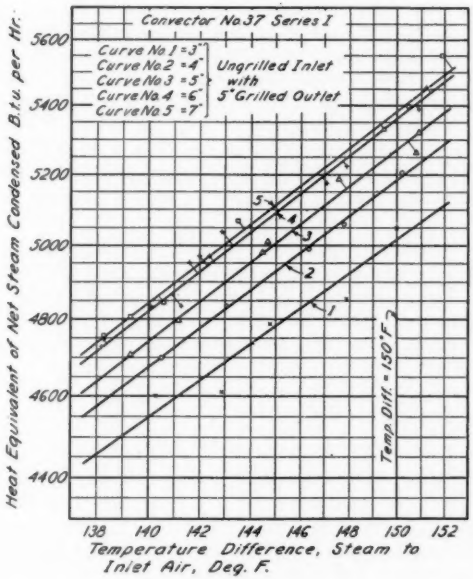


FIG. 5. TYPICAL PERFORMANCE CURVES FOR ONE SERIES OF TESTS

the standard temperature difference of 150 F. The first method consists of running a test under as nearly standard conditions as possible, and then correcting the heat equivalent of the steam condensed, making use of the formula:

$$H_s = H_t \left[\frac{150}{t_s - t_i} \right]^n \qquad (1)$$

in which H_s = the heat output under standard conditions, Btu per hour.
H_t = the heat output under actual test conditions, Btu per hour.
t_s = temperature of saturated steam under test conditions, degree Fahrenheit.

[5] Loc. Cit. See Note 1.

t_1 = average temperature of inlet air under test conditions, degree Fahrenheit.
n = an exponent depending on the type of convector (average value of 1.5 recommended in the Code).

This method involves either determining by tests the value of n for the particular convector, or accepting the average value of 1.5 recommended in the revised Code as being sufficiently accurate for all practical purposes.

A second method consists of running a number of tests at different inlet air temperatures, and thus establishing a performance curve giving the heat outputs over a fairly wide range of temperature differences between steam and inlet air. The heat output with the standard temperature difference of 150 F can then be read directly from the curve. The second method was adopted for the purpose of this investigation, and a typical set of performance curves for Convector No. 37 is shown in Fig. 5. In this series of tests Convector No. 37 had a 5-in. grilled outlet which was used in connection with ungrilled inlets varying in width by 1-in. increments from a width of 3 in. to one of 7 in. Each curve represents the results obtained with one width of inlet.

The test data shown in Fig. 5 were plotted to logarithmic scales, and hence the slopes of the resulting straight lines represent corresponding values of n in Equation (1). The values of n obtained from all of the test data resulting from this investigation varied between the limits of 1.43 and 1.69, and the average value agreed reasonably well with that of 1.5 recommended in the ASHVE Code.

Effect of Width of Inlet

A set of performance curves similar to that shown in Fig. 5 was developed for each test series on each of the three convectors listed in Fig. 4. For any given series of tests on each convector the heat output at the standard temperature difference of 150 F was read from one of these curves and plotted against the corresponding width of inlet or outlet. The heat outputs obtained for the three convectors with different widths of inlets, both grilled and ungrilled, are shown in Fig. 6. These curves present the results from Series I run with ungrilled inlets and Series II run with grilled inlets, and each point represents a heat output corresponding to a 150 F temperature difference as read from a curve similar to those shown in Fig. 5.

The curves in Fig. 6 show distinctly that the heat output of each convector was improved as the width of the inlet was increased. The heat output did not attain a definite maximum, but continued to increase until the width of the inlet became equal to the distance between the bottom of the heating unit and the floor. This distance constituted the physical limit beyond which it was not possible to increase the width of the inlet.

In every case, with inlets less than 5½ in. in width, the heat output obtained with the ungrilled inlet was greater than that obtained with the grilled inlet. Furthermore, when the width was increased beyond 5½ in., the heat output obtained with the grilled inlet approached that obtained with the ungrilled inlet, and in the case of Convector No. 35 the two heat outputs became equal with a width of inlet of approximately 5¾ in. It may also be observed that in

the case of the ungrilled inlets used with Convectors Nos. 35 and 37 the increase in heat output was not very marked after a width of approximately 5 in. had been reached, while in the case of the grilled inlets a marked increase occurred even up to the attainable limit in width.

Undoubtedly the heat output is affected by the frictional resistance of the convector as a whole, and the best width of inlet is probably determined by the relation between the resistance of the inlet itself and that of the heating unit,

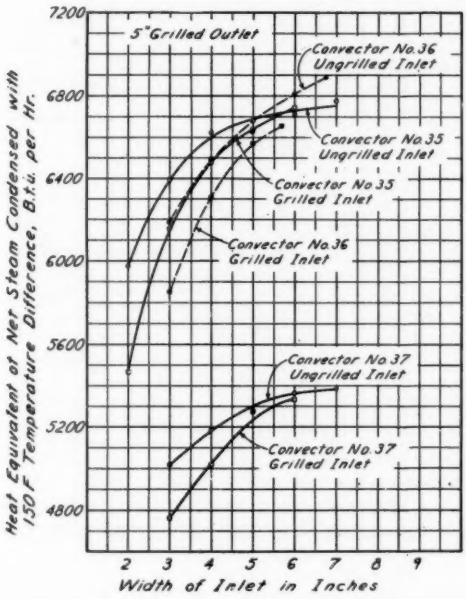


FIG. 6. HEAT OUTPUT OF CONVECTORS WITH DIFFERENT WIDTHS OF INLETS

since the latter resistance is usually large as compared with that of the cabinet and outlet. The resistance of the inlet is influenced by the free area, and that of the heating unit by both the free area and the height of the unit. With equal widths of inlets the free areas offered by the ungrilled were greater than those offered by the corresponding grilled inlets, and hence resulted in comparatively greater heat outputs. As the width of an ungrilled inlet increased beyond 5 in., however, the decrements in resistance became smaller as compared with the resistance of the heating unit, and the latter became the controlling factor. The effect of the resistance of the heating unit was further accentuated by the fact that any tendency for increased air velocity

was immediately reflected in increased resistance of the heating unit, thus tending to offset any gain resulting from further increase in the width of the inlet. The influence of the height of the heating unit is shown by the fact that, with the ungrilled inlets, in the cases of Convectors Nos. 35 and 37, which had 6-in. and 5¼-in. heating units respectively, a marked decrease in the increments of heat output with the increments of width occurred with widths greater than 5 in., while in the case of Convector No. 36, which had a heating

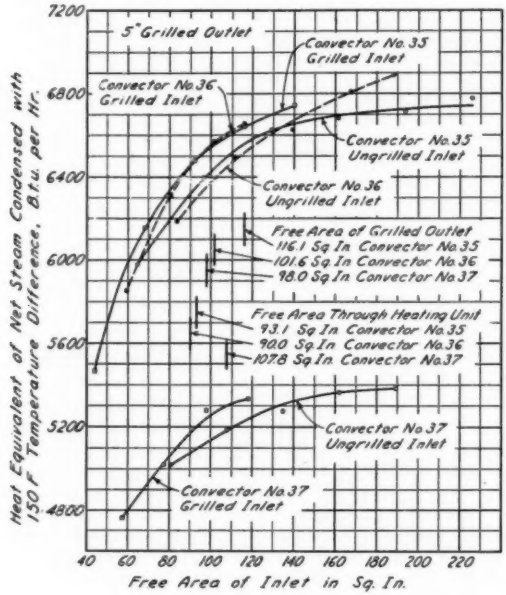


FIG. 7. HEAT OUTPUT OF CONVECTORS WITH INLETS HAVING DIFFERENT FREE AREAS

unit only 2 in. in height, well marked increments of heat output occurred when the widths of the inlets were increased above 5 in. In the case of the grilled inlets it was probably necessary to increase the free area up to a value comparable to the area of a 5-in. ungrilled inlet before the resistance of the heating unit became the controlling factor. Evidently this value was not reached with any of the limiting widths of grilled inlets attainable, because the increments of heat input showed no marked decrease as the widths of these inlets were increased above 5 in.

The results shown in Fig. 6 indicated that the relations between the free areas of the inlets and the free areas of the heating units, or more restricted portions of the convector, might be of more significance than the widths of the inlets. The heat outputs were therefore plotted against the free areas of the inlets,

and the results are shown in Fig. 7. These curves have the same general shape as those shown in Fig. 6, but the relative positions of the curves for the grilled and ungrilled inlets are reversed, owing to the fact that for given widths the free areas of the grilled inlets were considerably less than the areas of the ungrilled inlets. In the case of each convector, the heat output obtained with a given free area was greater with a grilled inlet than with an ungrilled inlet. The difference was small, never amounting to more than 3 per cent, and may have resulted from somewhat greater turbulence caused by the presence of the grille. Such increased turbulence would have a tendency to increase the heat transfer. The somewhat better performance with the grilled inlets may also have resulted from the fact that in the case of an ungrilled inlet a considerable portion of the air could flow around the upper edge of the opening and pass up through the front part of the heating unit, thus making this part of the unit more effective than the back, while with the grilled inlet the air entered over the whole face in jets and more of it was forced toward the back of the cabinet, thus increasing the effectiveness of the back part of the heating unit.

The curves shown in Fig. 7 tend to confirm the hypotheses made in connection with the discussion of Fig. 6. In the cases of Convectors Nos. 35 and 37, increasing the width of the ungrilled inlet beyond 4 or 5 in. resulted in no material increase in the heat output, indicating that with the greater widths of inlets the resistance of the heating unit became the dominant factor in influencing the heat output. The free areas through the heating units and the free areas of the outlets are shown in Fig. 7. The free area through the heating unit was regarded as the total unobstructed area of the inside of the cabinet at a horizontal section through the most obstructed plane in the heating unit. Each cabinet fitted the heating unit quite closely, thus preventing any bypassing of air around the sides and ends of the unit. It may be observed that the curve for Convector No. 35 with the ungrilled inlets tended to flatten after a free area of approximately 129 sq in. was reached, corresponding to a width of inlet of 4 in. For Convector No. 37 this flattening occurred at a free area of approximately 135 sq in., corresponding to a width of inlet of 5 in. Comparing these values with the free areas through the heating units, it is evident that for Convector No. 35 the critical width was accompanied by a ratio of free area of the inlet to free area of the heating unit of 1.39. The corresponding ratio for Convector No. 37 was 1.25. For Convector No. 36 no marked change in the slope of the curve appeared even with a ratio of 2.1, indicating that the critical width of ungrilled inlet was greater than 6¾ in. The critical ratio is apparently related to the height of the heating unit, since with Convectors Nos. 35 and 37, having 6-in. and 5¼-in. heating units respectively, ratios of the same order of magnitude were obtained, while with Convector No. 36, having only a 2-in. heating unit, the ratio was considerably greater. The correlation between the critical ratios and the heights of heating units was not exact, because undoubtedly factors other than height and free area, such as arrangement of fins, affect the frictional resistance of the heating unit.

It may be further observed from Fig. 7, that, in the case of grilled inlets, the curves also show some tendency to change slope with free areas of inlets of approximately 95 and 110 sq in. for Convectors Nos. 35 and 37 respectively. These both correspond to a ratio of free area of the grilled inlet to the free area of the heating unit of 1.02. In the case of Convector No. 36, with only a

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2-in. heating unit, the ratio was again greater, having a value of 1.17. The results shown in Fig. 7 therefore indicate that in the case of convectors having heating units approximately 6 in. in height, as in Convectors Nos. 35 and 37, the free area of an ungrilled inlet should be approximately 25 per cent in excess of the free area through the heating unit, and no appreciable gain in heat output can be obtained by increasing the free area of the inlet materially above 25 per cent in excess of the free area of the heating unit. In the case

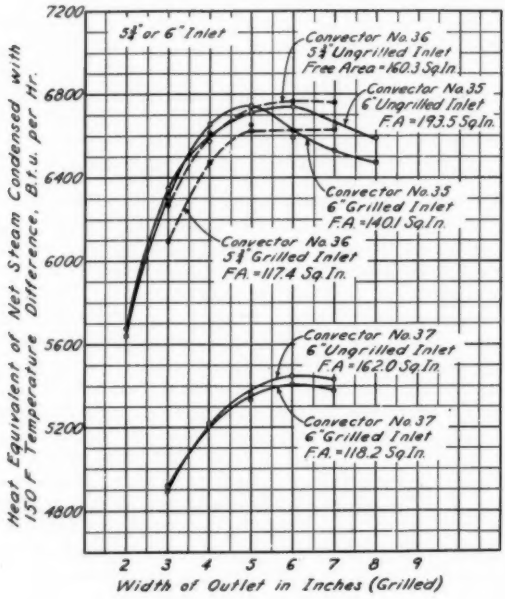

FIG. 8. HEAT OUTPUT OF CONVECTORS WITH DIFFERENT WIDTHS OF OUTLETS

of convectors having heating units of approximately 2 in. in height, the effective free area of an ungrilled inlet is limited only by the distance between the floor and the bottom of the heating unit. The free area of grilled inlets for all convectors should be at least equal to the free area of the heating unit.

Effect of Width of Outlet

The heat outputs obtained from Series III and Series IV, using a constant width of ungrilled or grilled inlet with different widths of grilled outlets, are shown in Fig. 8, which was developed by the same method as that employed for Fig. 6. The curves in Fig. 8 all present the same characteristics, in that as the width of outlet was increased the heat output also increased, but reached

a distinct maximum when the width of the outlet was 5 or 6 in. Increases above 5 or 6 in. in width were accompanied by more or less marked decreases in heat output. With Convectors Nos. 35 and 37 the substitution of the grilled inlet had no material effect on the heat output for widths of outlets up to 5 in., while with Convector 36 such substitution had a marked effect over the whole range.

Several explanations may be offered for the tendency of the heat output to reach a maximum with successive increases in the width of the outlet. The width was increased by lowering the bottom edge of the outlet opening, and with increasing widths of outlets this procedure may have reduced the effective stack height, and thus somewhat reduced the flow of air through the heating unit. It is rather difficult, however, to determine just what does constitute effective stack height in any given case.

In any stack with a front outlet similar to a convector cabinet the air tends to crowd to the top before making the turn out of the outlet. This condition results in high air velocities toward the top of the outlet and the upper portion becomes more effective than the lower in passing air. As the bottom of the outlet is lowered, the high velocity portion of the air stream tends to remain approximately the same width, and the lower portion of the opening becomes less and less effective. Finally a width of outlet is reached which is just equal to the width of the air stream, and if the width is now increased by lowering the bottom edge, air from the room will enter the lower part of the opening. This air will be drawn into the main air stream and be discharged through the upper portion of the opening without having passed through the heating unit. This interference tends to reduce the flow of air and thus to reduce the heat output of the convector. Hence, beyond a certain width, the increases in width would merely permit more air to be drawn in at the bottom of the opening, and progressive increases in width would be accompanied by decreases in the heat output.

The heat outputs obtained from Series III and IV were plotted against the free areas of the outlets, and the results are shown in Fig. 9. Since only grilled outlets were used, there was a fixed relationship between the free area and the width of the grille. Hence, the curves in Fig. 9 present the same characteristics as those shown in Fig. 8. In addition, however, the free areas of the inlets and of the heating units are shown in Fig. 9. It is probable that the heat output of a convector is influenced by the relative resistances of the outlet and the heating unit, just as it was shown to be by the relative resistances of the inlet and the heating unit. Some indication of this condition is shown by the curve representing the results obtained from Convector No. 35 when used with a grilled inlet. This convector had a 6-in. heating unit, and when used with a grilled inlet it probably offered the greatest combined resistance of any of the convectors. In this case the reduction in heat output with successive increments in the width of the outlet in excess of 5 in. or free area in excess of 116 sq in. occurred at a greater rate than it did in the case of any other arrangement. Further indication is given by the curves for Convector No. 36. This convector had a heating unit only 2 in. in height, and offered the least resistance of any of the convectors. In this case the substitution of a grilled inlet for an ungrilled inlet resulted in a marked reduction in heat

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output over the whole range, indicating that the resistance of the heating unit was not sufficient to become the controlling factor and thus allowed the additional resistance offered by the grille to be directly reflected in a reduction in heat output. In Convectors Nos. 35 and 37 the resistances of the heating units were sufficiently high to prevent the small additional resistance of the grille at the inlet from being directly reflected in reductions in the heat outputs.

A study of the ratios of the free areas of the outlets to the free areas of the heating units at maximum heat output as shown in Fig. 9 failed to prove

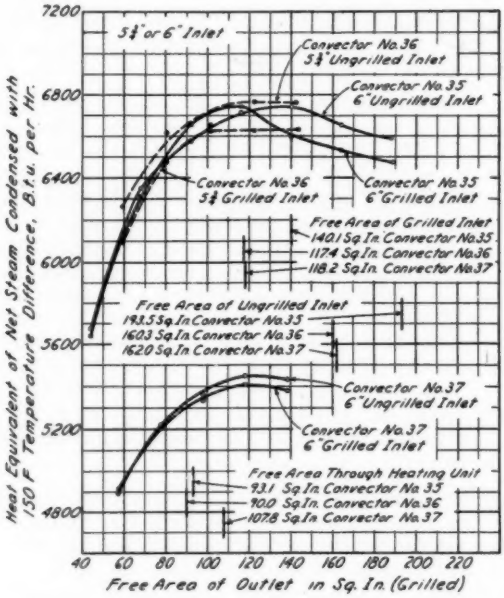

FIG. 9. HEAT OUTPUT OF CONVECTORS WITH OUTLETS HAVING DIFFERENT FREE AREAS

any very definite correlation between the areas themselves or between the areas and the heights of the heating units; further than that the free area of the outlet should not be less than 10 per cent and probably not greater than 30 per cent in excess of the free area of the heating unit. The one exception proved to be Convector No. 35 when used with an ungrilled inlet, in which case the ratio for maximum heat output was 1.45. The ratios of the free areas of the inlets used with the three convectors to the free areas of the heating units varied from 1.1 to 1.5 for the grilled inlets, and from 1.5 to 2.1 for the ungrilled inlets. Hence, the resistances of the inlets used with the different widths of outlets did not have any material influence on the maximum heat

output, and the ratios of the free areas of the outlets to the free areas of the inlets do not seem to have any particular significance in this study.

Effect of Reduced Length of Inlet

In order to conceal piping connections it is more or less common practice for the length of the inlet to be from 6 to 8 in. shorter than the overall length of the convector cabinet. Studies were made to determine whether this practice entailed a sacrifice in heat output, and the results, plotted to logarithmic scales, are shown in the left half of Fig. 10. Details of the different arrangements are shown in the insets at the top of the figure. Curve No. 35*a* shows the heat outputs obtained from Convector No. 35 with an inlet 32¼ in. in length and 6 in. in width, while Curve No. 35*b* shows the heat outputs obtained when the inlet was shortened 6 in. by reducing it 3 in. at each end. Curves Nos. 35*c* and 35*d* respectively show similar results obtained with an inlet 4 in. in width. It may be observed that with the inlet 6 in. in width no sacrifice in the heat output resulted from the reduction in length. The small apparent gain indicated probably represents the limit of the ability to exactly duplicate tests rather than an actual increase in heat output. In the case of the 4-in. inlet, however, the reduction in the length of the inlet was accompanied by a reduction of approximately 1.5 per cent in heat output at the standard temperature difference of 150 F. It is probable that the same criterion should be applied to the reduction in length as that developed for the reduction in width of inlet, namely, that the free area of an ungrilled inlet should be at least 25 per cent in excess of the free area through the heating unit.

Effect of Directing Vanes

The curves in the right half of Fig. 10 show the results obtained when vanes of different types were installed at the inlet or outlet of a convector cabinet. Convector No. 35, having a 6-in. ungrilled inlet and a 5-in. grilled outlet, was used as a base construction and the details of the modifying vanes are shown in the various insets. The basic cabinet is designated as Convector No. 35*a* and the corresponding curve of heat outputs as Curve No. 35*a*. The addition of the vane in the top of the cabinet, Convector No. 35*e*, resulted in a slight improvement in the heat output, as shown by Curve No. 35*e*. This improvement amounted to approximately 1½ per cent at the standard temperature difference of 150 F. The use of a vane at the center in addition to one at the top, Convector No. 35*f* and corresponding result Curve No. 35*f*, resulted in a gain of approximately 4 per cent at the standard temperature difference of 150 F, as compared with the heat output of Convector No. 35*a* used without vanes. The removal of the vane at the top, thus converting Convector No. 35*f* into Convector No. 35*h*, did not result in an appreciable decrease in heat output as compared with that obtained with the two vanes in Convector No. 35*f*. The addition of a vane at the inlet, converting Convector No. 35*f* into Convector No. 35*g*, resulted in a slight decrease in heat output, indicating that nothing is to be gained by using vanes at the inlet of a convector.

The conditions shown by Curves Nos. 35*a* to 35*h* can be readily explained by the character of the air flow. In the case of a front outlet similar to the one in Convector No. 35*a*, the air tends to crowd to the top and the lower

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portion of the outlet is only partially effective in passing air. The banking of the air in the upper corner forms a more or less dead air space that tends to serve as a directional surface without the necessity for a vane. Using a vane in this location may give the air flow slightly better direction, but does not correct the ineffectiveness of the lower part of the face of the outlet. Hence no great improvement is effected by the addition of this vane, as shown by the comparison between Curves Nos. 35*a* and 35*e*, and Curves Nos. 35*h* and 35*f*. When the vane is placed at the center, however, the air flow is directed more uniformly over the whole face of the outlet, and the lower part becomes effective. Hence, this arrangement offers the best possibility for material improvement over the outlet with no vanes, as substantiated by Curves

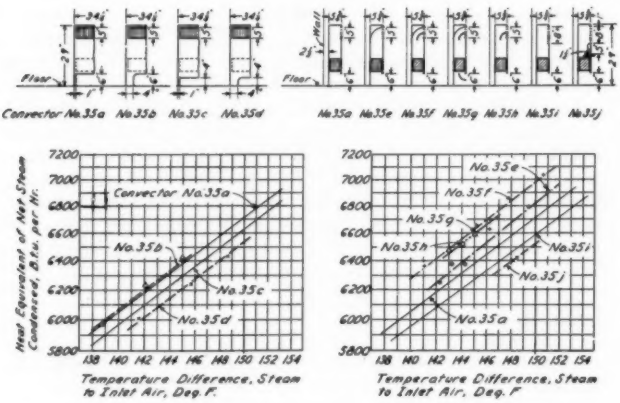

FIG. 10. PERFORMANCE CURVES FOR CONVECTOR NO. 35 USING REDUCED LENGTH OF INLET AND DIFFERENT ARRANGEMENTS OF VANES AT INLET AND OUTLET

Nos. 35*h* and 35*a*. Again it may be observed by comparing Curves Nos. 35*h* and 35*f* that the addition of the vane at the top of the convector cabinet resulted in no material gain in heat output. The air flow into an inlet is more or less uniform over the whole face. Hence, the addition of a vane at the inlet is more liable to act as a detriment by introducing additional friction than it is to serve as a guide to give more uniform air flow over the heating unit.

The resistance to air flow around a bend or out of a front or side outlet is to a much greater extent determined by turbulence existing at the inside edge or corner, than it is by conditions existing at the outside corner. Conditions at the inside corner can sometimes be improved by streamlining or creating a venturi throat on the down-stream side of the turn. An attempt was made to improve the heat output obtained with the 8-in. width of outlet used with Convector No. 35 by the use of such a venturi throat. The details of this throat are shown in the inset in the right half of Fig. 10. It should be noted that it was not possible to place this vane on the down-stream side,

since this would have located it outside of the outlet. Hence, it was placed on the up-stream side. In the case of elbows this construction has been shown to have some merit, but it is not nearly as effective as a venturi on the down-stream side. The basic curve for Convector No. 35 with an 8-in. outlet is shown as Curve No. 35*i* in Fig. 10. The use of the venturi throat resulted in a net loss in heat output, as shown by Curve No. 35*j*. Hence, the use of such an arrangement is not to be recommended.

Effect of Different Locations of Inlets and Outlets

The cabinet furnished with Convector No. 31*a* had two inlets, one in the front and one in the back of the cabinet. With the back inlet blocked this convector was designated as Convector No. 31. Details of these arrangements are shown in the insets in the lower part of Fig. 11. The performance curves plotted to logarithmic scales for Convectors Nos. 31 and 31*a* in Fig. 11 show that when the back inlet was blocked the heat output at the standard temperature difference of 150 F was reduced approximately 4 per cent. Hence, in this case the use of the double inlet could be justified. It should be noted, however, that in this particular convector the front inlet was only 3½ in. in width, and the extra area could probably just as well have been obtained by making this inlet wider instead of using a second inlet. A double inlet, however, might be advantageous in the case of a convector with the heating unit located close to the floor.

Studies were made on two convectors to determine whether any improvement in heat output could be effected by using a top outlet instead of a front outlet having the same free area. Convector No. 33 furnished with a commercial cabinet and Convector No. 35*c* equipped with a special cabinet were used for this purpose, and with the modified cabinets having top outlets were designated as Convectors Nos. 33*a* and 35*k* respectively. The details of these arrangements are shown in the insets in the upper part of Fig. 11. The results shown in Fig. 11 indicate that a marked improvement in heat output resulted from the use of a cabinet having a top outlet instead of a front outlet. In the case of Convector No. 33 the gain in heat output effected by the use of the top outlet was 14.5 per cent at the standard temperature difference of 150 F, while in the case of Convector No. 35*c* the gain was 9.5 per cent. Undoubtedly the amount of gain depends on the relative resistances of the heating unit and the original front outlet, and on the relative active stack heights involved. Sufficient data were not available, however, to permit any accurate evaluation of the effects of these additional factors.

Effect of a Humidifying Pan on the Heat Output

The construction of a convector suggested that it might be modified to serve as a device for both heating and humidifying the air. Convector No. 11 was therefore adapted to humidify the air by installing a water pan having a false bottom. The space between bottoms was connected to the steam space in the heating unit by tubes at the ends, as shown in the inset in Fig. 12. The original 4¾-in. outlet was increased to a width of 8 in. and the modified convector was designated as Convector No. 11*a*.

The performance curve for the original convector, No. 11, is shown as the lowest curve in Fig. 12. The total heat outputs, corresponding to the total

steam condensed by both the heating unit and the water pan, are shown for Convector No. 11*a* by the top curve in Fig. 12. Apparently this total heat output was more or less independent of the temperature difference between the steam and the air entering the inlet of the convector. The total amount of water evaporated remained practically constant at a value of 2.0 lb per hour, or an evaporation of 2.6 lb per square foot of surface per hour. The

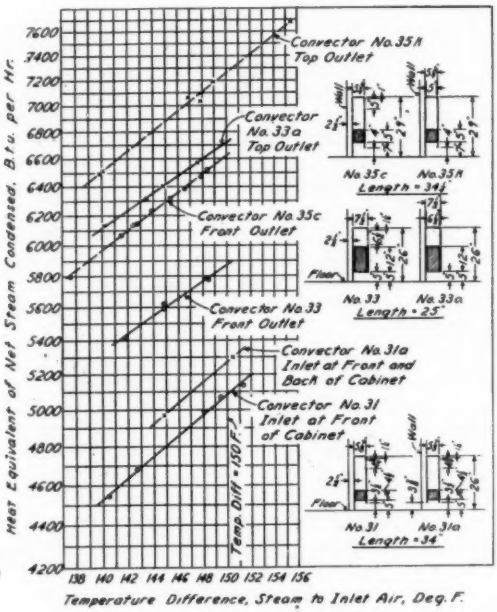

FIG. 11. PERFORMANCE CURVES FOR CONVECTORS NOS. 31, 33 AND 35 WITH DIFFERENT ARRANGEMENTS OF INLETS AND OUTLETS

water temperature varied from approximately 190 F at a temperature difference of 137 F between steam and entering air to approximately 180 F at a temperature difference of 150 F. Based on one air change per hour, with a temperature of 70 F indoors and zero outdoors, the continuous evaporation of 2.0 lb per hour is sufficient to maintain approximately 5800 cu ft of space at a relative humidity of 35 per cent. Hence, Convector No. 11*a* may be regarded as a very effective humidifier. The air leaving the convector was not saturated and no evidence of moisture condensing on the cabinet was observed.

The effectiveness of Convector No. 11*a* as a heating device, including heat added to the air and heat radiated to the room, was obtained by subtracting

the latent heat of evaporation at the temperature of the water in the pan from the heat equivalent of the total steam condensed. The heat outputs thus obtained are shown by the intermediate curve in Fig. 12. It may be observed that, at the standard temperature difference of 150 F, the effective heat output of Convector No. 11*a*, with the water pan, represented an increase of approximately 8 per cent over the heat output of the original Convector No. 11, without the water pan. This increase in heat output can probably be explained by the fact that the superficial area of the pan containing water at from 180 F to 190 F was at a higher temperature than that of the air passing over it. It thus becam: effective heating surface in addition to the

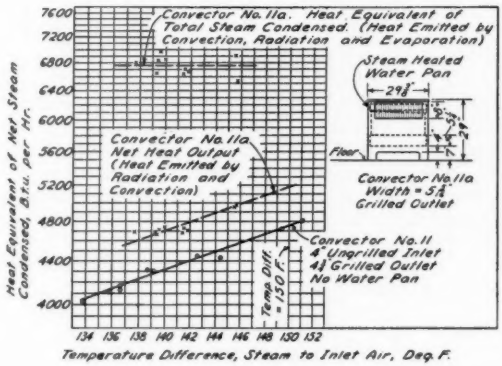
Convector No. 11a. Heat Equivalent of Total Steam Condensed. (Heat Emitted by Convection, Radiation and Evaporation)
Steam Heated Water Pan
29"
Convector No. 11a Net Heat Output (Heat Emitted by Radiation and Convection)
Floor
Convector No. 11a Width = 5A Grilled Outlet
Convector No. 11 4" Ungrilled Inlet 4½ Grilled Outlet No Water Pan
Temp. Diff. = 150° F.
Heat Equivalent of Net Steam Condensed, B.t.u. per Hr.
7600
7200
6800
6400
6000
5600
5200
4800
4400
4000
134 136 138 140 142 144 146 148 150 152
Temperature Difference, Steam to Inlet Air, Deg. F.

FIG. 12. EFFECT OF INSTALLING STEAM HEATED WATER PAN AT OUTLET OF CONVECTOR NO 11

normal heating surface of the heating unit, the bottom of the pan and the connecting pipes which were at steam temperature.

CONCLUSIONS

The following conclusions may be drawn as applying to the data obtained from these tests:

(1) From the standpoint of heat output the free area of a grilled inlet should be at least equal to, and of an ungrilled inlet should be approximately 25 per cent in excess of, the free area through the heating unit.

(2) For convectors having heating units 5 or 6 in. in height, or those having heating units with comparatively high frictional resistance, no material gain in heat output can be obtained by increasing the free area of an ungrilled inlet to more than 25 per cent in excess of the free area through the heating unit.

(3) For maximum heat output the free area of the outlet of a convector should not

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be less than 10 per cent and probably not greater than 30 per cent in excess of the free area through the heating unit.

(4) The most effective position for a directing vane at the outlet of a convector cabinet is in a central location with respect to both the cabinet and the outlet face.

(5) A directing vane located at the top of a convector cabinet is effective to a very small extent.

(6) Directing vanes at the inlet to a convector cabinet are not effective and may prove to be detrimental by introducing additional friction.

(7) The heat output of a convector having a cabinet with a top outlet may be from 9.5 to 14.5 per cent greater than the heat output from the same convector with a cabinet having a front outlet with the same free area as that of the top outlet.

(8) A convector may be converted into an effective humidifier without loss in effectiveness as a heating device by installing in the top of the cabinet a water pan provided with a steam space connected to steam headers in the heating unit.

ACKNOWLEDGMENTS

The data presented in this paper were obtained in connection with an investigation conducted by the Engineering Experiment Station of the University of Illinois, of which M. L. Enger, dean of the College of Engineering, is the director. The investigation was conducted in the Department of Mechanical Engineering, of which O. A. Leutwiler, professor of Mechanical Engineering Design, is the head. The results will ultimately comprise part of a bulletin of the Engineering Experiment Station.

DISCUSSION

W. A. ROWE (WRITTEN): In the case of convectors having a cabinet height the same as used in this series of tests, the draft or stack effect, is of the order of 0.005 in. water gage. This pressure difference must take care of the resistance to air flow of the heating element, of the inlet and outlet losses, and the velocity head. Thus we are dealing with very minute quantities, and the problem of getting capacity increases of a material amount is exceedingly difficult.

The design of the heating element is important, but apparently not so much so as the enclosing cabinet and the proportions of the air inlet and outlet openings, which is the field covered by this paper. And changes in the heating element to be worthwhile may very likely involve an increase in the cost of the unit; whereas, in the case of the inlet and outlet openings it may only be a matter of a changed dimension or two. It would seem that this investigation has started at the point where greatest returns are likely to be realized.

It should be noted that the conclusions arrived at are based, for each type of unit tested, on but one depth of unit and one cabinet height. To be complete such determinations should embrace every depth of unit and cabinet height listed. The optimum proportions thus established should be adhered to in case the cabinet is designed or made by others than the manufacturer of the convector.

It is to be hoped that the good work done so far will be continued in the future along parallel lines.

R. H. NORRIS[6] (WRITTEN): Results of much significance and practical value are provided and the authors deserve commendation. There is, however, one feature of

[6] General Engineering Laboratory, General Electric Co., Schenectady, N. Y.

the results on which further information would be desirable, namely, the relative flow resistances of the different heating units used in the tests.

The authors consider the vertical height of the heating unit elements, and the minimum free area of the air passages through these elements as the factors which affect the resistance to air flow of the unit. It seems likely that there may be other equally important factors affecting this air flow resistance, in particular the width, or the equivalent diameter, of the individual air passages between the heating unit fins, and the per cent variation of free area of these air passages caused by the passage of the air around the steam tubes. Quantitative data on these additional factors and a discussion of their significance would be desirable.

The importance of these additional factors arises from the fact that the air flow in the heating units will normally be laminar, not turbulent.

Regardless of the kind of flow, the flow resistance or pressure loss is made up of two components: the dynamic losses introduced by changes in velocity, and the losses due to friction over the surfaces.

With turbulent flow, the major losses would be dynamic losses, and since these vary as the square of the air velocity and are independent of diameter, for a given velocity, neglect of the effect of diameter, and, of the increase of the free area above the minimum value, would probably be permissible.

With laminar flow, however, the friction pressure loss is likely to be a larger fraction of the total. This friction pressure loss is proportional only to the first power of the velocity, and inversely proportional to the *square* of the equivalent diameter. Hence, if much of the air passage free area is considerably greater than the minimum, the flow resistance would be appreciably less than if the free area remains substantially constant.

The effect of diameter is of particular interest in comparing convectors Nos. 35 and 36. Since the equivalent diameter is roughly proportional to the distance between adjacent fins, and since this distance between fins appears to be about two or three times as large in convector No. 35 as it is in No. 36, the friction component of pressure loss might well be expected to be less for No. 35 than for No. 36, even after allowing for the effect of the differences in height of the two units. The fact that the minimum free areas are substantially the same, and that the test results indicate that the *total* flow resistances have the opposite relation (a greater resistance for No. 35 than for No. 36) indicates that the differences in the flow resistance have not yet been satisfactorily explained. Perhaps this explanation may involve the different degrees of variation of the free area in the two convectors.

JOHN MCELGIN (WRITTEN): The authors are to be commended for directing the attention of the Society to a subject that is too often ignored in convector application and design. While a loss in capacity could be naturally anticipated for a restricted inlet, the fact that an oversized or incorrectly shaped outlet will also inflict a loss in capacity is not generally known.

It would have been of interest had the authors investigated the influence of length of outlet upon the capacity top outlet convector. It has been our experience that in this case the physical length of the outlet in relation to both active convector length and to cabinet length is of first importance.

If the length of the outlet is much in excess of the active or finned length of the convector we find that re-entry of air will invariably occur at the ends and a loss in capacity will result.

There often occurs in actual application the case where the length of the cabinet exceeds the convector length in order to provide space for enclosed valves and traps.

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Here we find it desirable to not only limit the length of the outlet to the active convector length but in addition to install interior partitions to confine the heated air within the active convector length.

ERNEST SZEKELY and C. H. RANDOLPH (WRITTEN): Essentially the authors have done a very laudable job in working out a procedure and producing test data which seems of practical value. The conclusions are clear and serviceable as rendered.

There is no intimation in the paper whether or not it is the intention of the authors to continue this investigation with convectors, but if they have such plans, we would like to submit the following suggestions:

1. Experiments studying the effect of stack height.
2. Experiments with mediums other than steam.
3. Studying the effect of baffling or artificial turbulences both inside or outside of convector.

1. The paper as it is now written is based on tests with a cabinet having a constant height of 29 in. It may be desirable to know how convector performance is affected when placed in a low cabinet say only 22 in. high and again it may be of practical value to know the performance of convector in relatively high stacks (7 to 8 ft high).

2. In addition to the results now obtained, it would be desirable to know the performance of convectors circulating hot water, cold water, and refrigerants.

3. Inasmuch as the paper, no doubt, had for its purpose the determination of the best working conditions for convectors, it seems it would be also desirable to know what improvements could be made by introducing deflector plates in the air side and some means provided to create artificial turbulences on the fluid side with the idea of studying the influence of greater scrubbing effect on primary surfaces.

These may not have been the ideas of the authors but if time permits and if funds are available, valuable data could be secured by the extension of the work along the lines suggested. Item No. 3 would be of particular interest, because when performance is based on gravity it is hard to estimate by speculation whether or not changes affect results in the plus or minus direction.

H. M. HART: This is an excellent contribution that will be very useful in the industry. I would like to know whether further tests are being conducted with varying temperature differences such as are commonly used in hot water heating. No information is now available on this.

E. HOLT GURNEY: I would imagine that the first curves shown, which had to do with the size of the inlet for air, would go even further apart, if the convector had been a hot water convector with, say, an average temperature through the element of 180 F. Do you agree?

PROF. M. K. FAHNESTOCK: We did none of these tests with hot water, so I am unable to answer your question.

MR. GURNEY: Would you be willing to give an educated guess?

PROFESSOR FAHNESTOCK: Your assumption might well be right. There seems reason to suppose that this might be the case; but remember, we have no data.

H. J. YOUNG: Using a 2¼ in. deep fin on the copper core, if you doubled or rather stacked another core on this, the second core would only give approximately 16 per cent more heat.

No. 1134

A THEORY COVERING THE TRANSFER OF VAPOR THROUGH MATERIALS

By FRANK B. ROWLEY,* (*MEMBER*), MINNEAPOLIS, MINN.

THERE has been much speculation about the theory relating to the transfer of vapor through materials and the application of the theory to building construction. For convenience it has often been assumed that the laws for vapor transmission are similar in form to those governing the flow of heat through the walls of a building, and that coefficients of vapor transmittance may be developed for materials or combinations of materials which may be applied in the same manner as coefficients of heat transmission. In making this analogy it is assumed that the difference in vapor pressures between two parts of a structure is the motive force which causes the flow of vapor and that the amount of vapor transmitted is directly proportional to the difference in vapor pressure and inversely proportional to the vapor resistance between the two parts of the wall. The over-all vapor resistance of a built-up wall section is assumed to be equal to the sum of the vapor resistances of its component parts. If the analogy were complete a vapor transmittance coefficient would be defined as the quantity of vapor transmitted per unit cross-sectional area, per unit time, per unit difference in vapor pressure along the path of transmittance, and there would be coefficients for surfaces, air spaces, homogeneous materials, and combinations of materials.

Before accepting a complete analogy between the two problems an analysis should be made to determine those elements which are similar and those which may be conflicting. Heat is a form of energy having no physical properties, and unless it is changed from sensible to latent, or vice versa, it will be transferred through a wall without change of state. Vapor is a substance with physical properties and in the course of its transfer through a structure may change its state several times. Heat may be transmitted by radiation, conduction, and convection. Vapor may be transmitted by molecular diffusion and convection, and the condensed vapors may be transmitted by capillarity or other means.

Water vapor in the air is steam and has the same physical properties that it would have if the air were not present. If air at a given temperature is saturated with water vapor, the density of the vapor in the air is the same as the density of saturated steam at the given temperature, and the pressure of

* Director, Engineering Experiment Station, University of Minnesota.

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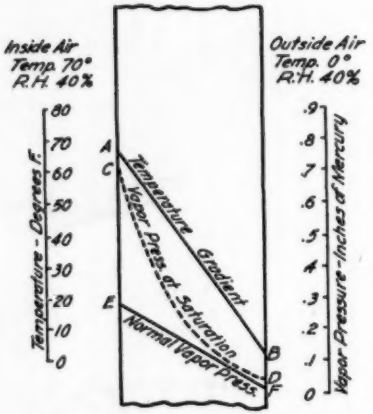

FIG. 1. WALL BUILT OF NON-HYGROSCOPIC, HOMOGENEOUS MATERIAL PERMEABLE TO WATER VAPOR. *C-D* REPRESENTS MAXIMUM VAPOR PRESSURE THAT CAN EXIST IN WALL WITHOUT CONDENSATION. *E-F* REPRESENTS NORMAL VAPOR PRESSURE ESTABLISHED BY AIR CONDITIONS ON TWO SIDES OF WALL. CONDENSATION WILL NOT OCCUR

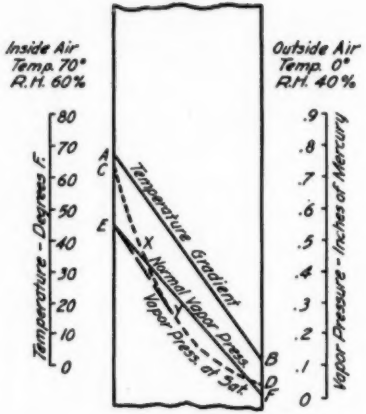

FIG. 2. WALL BUILT OF NON-HYGROSCOPIC, HOMOGENEOUS MATERIAL PERMEABLE TO WATER VAPOR. *E-F* REPRESENTS NORMAL VAPOR PRESSURE ESTABLISHED BY ATMOSPHERIC CONDITIONS ON TWO SIDES OF WALL. SINCE *E-F* CROSSES MAXIMUM VAPOR PRESSURE *C-D* CONDENSATION WILL OCCUR. *E-Y-F* REPRESENTS ACTUAL VAPOR PRESSURE ESTABLISHED IN WALL

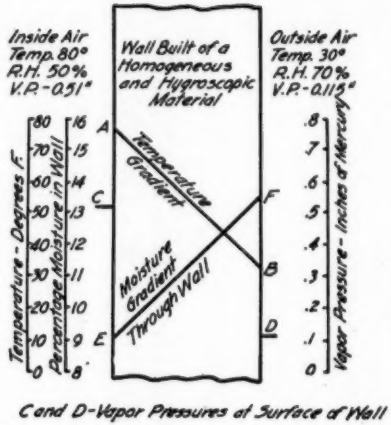

FIG. 3. WALL BUILT OF HOMOGENEOUS, HYGROSCOPIC MATERIAL IMPERMEABLE TO WATER VAPOR. MOISTURE WILL TRAVEL THROUGH WALL IN REVERSE DIRECTION TO VAPOR PRESSURE DROP BETWEEN TWO SURFACES OF WALL

the vapor in the air corresponds to the boiling pressure of water at the given temperature. If the temperature is raised the vapor is superheated, but the vapor pressure is not changed. If the temperature is lowered, some of the vapor is condensed and the vapor pressure and density are reduced to values corresponding to saturation at the new temperature.

The method by which water vapor travels from one point to another depends partly upon the material through which it travels and partly upon the temperature along its path. Insofar as its transfer through building construction is concerned the materials through which it travels may be classified as air and solids. The conditions of air which seem to be of greatest importance are temperature, pressure, and air movement. The properties of the solids which seem to have the greatest significance are permeability to gases and power to absorb and condense water vapor from the air. These two properties of solid materials together with temperature largely determine the state of the moisture as it travels through them.

If the temperature of air is above the dew-point temperature of vapor the vapor may be transmitted through the air by turbulence and by molecular diffusion. The transfer of vapor by convection currents is similar to the transfer of heat by convection currents. The resistance to the flow of vapor through air by molecular diffusion is a function of density and temperature of the air through which it passes. For a given temperature and density of air vapor mixture the rate of vapor travel by molecular diffusion between two points may be considered as directly proportional to the difference in vapor pressure and inversely proportional to the distance between the points. Thus under ordinary conditions it may be substantially correct to assume that the laws governing the rate of vapor travel through air by either convection currents or molecular diffusion are similar in form to those governing the flow of heat through air by conduction and convection.

When vapor passes through a solid material there are at least three types of materials which seem to be of importance. First, those materials which are permeable to air or gas and which will not absorb water vapors; second, those materials which are impermeable to gas but which will absorb water vapor; and third, those materials which are permeable to gas and which will also absorb water vapor. Temperature is also an important factor.

If a non-hygroscopic material is permeable to water vapor and its temperature at all parts is above the dew-point temperature of the vapor in contact with those parts, the vapor will travel through the material without change in state. If the static pressure on the two sides of the material is balanced, the vapor transmittance should be by molecular action and the rate of transfer should be proportional to the vapor pressure difference and inversely proportional to the distance of travel. The unit vapor resistance of the material should depend partly upon arrangement of cells or open spaces through which the vapor may travel, and it should be higher than the unit resistance for still air. If the temperature at some point along the path is below the dew-point temperature of the vapor at that point, condensation may occur, forming either water or ice depending upon the temperature. The uniform transfer of vapor will be disturbed, new vapor pressure gradients will be established, and there will be a tendency to accumulate moisture within the material.

If a material which is impermeable to water vapor is hygroscopic, it will

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condense water vapor and establish a moisture equilibrium when in contact with air vapor mixtures. The percentage of moisture absorbed by the material will depend somewhat on the temperature, but largely upon the relative humidity of the air with which it is in contact. The transmission of moisture through materials of this character will be by capillarity or some similar process and will not depend upon vapor pressure difference. Under certain conditions it appears that vapor may be absorbed from a mixture of low vapor pressure, transmitted through the material, and delivered to a mixture of high vapor pressure. If the temperature of either surface of the material is below the dew-point temperature of the vapor in contact with it, some of the vapor will be condensed forming either free water or frost, depending upon the temperature. Free moisture or water may disturb the normal moisture equilibrium which would have been established by the water vapor in contact with the material and thus change the rate, and perhaps direction, of moisture flow through the material.

If a material is hygroscopic and permeable to water vapor, it may transmit water vapor and free moisture, the net result depending upon several factors. For instance, a condition might be set up in which vapor would be transmitted in one direction due to vapor pressure differences and the permeability of the material, and moisture would be transmitted in the opposite direction due to the hygroscopic properties of the material and the relative percentages of vapor saturation on the two sides of the material.

The wall of Fig. 1 is built of a homogeneous, non-hygroscopic material which is permeable to water vapor. The air in contact with the left hand surface of the wall is at 70 F and 40 per cent relative humidity, and that in contact with the right hand side is at 0 F and 40 per cent relative humidity. The temperature gradient line through the wall is plotted to the temperature scale shown at the left, and the vapor pressure lines are drawn to the vapor pressure scale shown at the right. In each instance a reasonable drop in temperature and vapor pressure is indicated between the surface of the material and vapor in contact with this surface.

Under the conditions given the line *A-B*, plotted to the temperature scale, represents the temperature gradient through the wall. The curved line *C-D*, plotted to the vapor pressure scale, represents the maximum vapor pressure that it would be possible to carry at any point within the wall without condensation. In other words any point on this curve gives vapor pressure at saturation for the temperature of the material in the wall at that point. The straight line, *E-F*, is the normal vapor pressure gradient established in the wall due to the vapor pressures on each side of the wall. In this case the line *E-F* does not cross the curved line *C-D*, indicating that the temperatures at all points in the wall are above the dew-point temperature of vapor at corresponding points. The vapor should thus pass from *E* to *F* through the wall without condensing within the wall.

The wall shown in Fig. 2 is identical in every respect with that shown in Fig. 1, and the conditions as to surrounding temperatures and vapor pressures are the same with the exception that the relative humidity of the air on the left hand surface of the wall has been raised from 40 to 60 per cent, thus increasing the pressure of the vapor in contact with this surface. In this case the vapor pressure line, *E-F*, which would be established through the wall

due to vapor pressure conditions in contact with the two surfaces will be much steeper than that for Fig. 1, and as shown will cross the limiting vapor pressure line *C-D* at some point *X* within the wall. This means that the temperature of the material at *X* will be below the normal dew-point temperature of the vapor at this point and condensation should take place. Since the temperature of the material in the wall at points immediately beyond *X* is such that the maximum vapor pressures are below those which would normally be established, the rate of vapor travel will be accelerated and new vapor pressure gradients will be established through the wall. If the straight line, *E-Y*, is drawn tangent to the curve, *C-D*, at *Y*, it seems probable that *E-Y* would represent the vapor pressure gradient established in the wall up to the point *Y*. From this point on the maximum vapor pressure gradient for each successive point would be established by a tangent to the curved line, *C-D*, and when the slope of this tangent is less than the slope of the line *E-F*, the rate of vapor travel would be less than the normal rate. The rate at which moisture enters the wall would thus be increased and the rate at which it leaves would be decreased from the normal, and there would be an accumulation of moisture within the wall from the point *Y* to the outer surface. Since in this case the interior temperatures of the wall would be below 32 F, frost or ice would be formed.

The wall of Fig. 3 is built of a homogeneous hygroscopic material which is impermeable to water vapor. The air on the left hand side is at 80 F and 50 per cent relative humidity, and that on the right hand side is at 30 F and 70 per cent relative humidity. The normal temperature gradient through the wall is shown by the straight line *A-B* plotted to the temperature scale at the left. The pressure of the water vapor in contact with the two surfaces is shown by the two short heavy lines, *C-D*, and plotted to the vapor pressure scale on the right. This represents the vapor pressure difference on the two sides of the wall, but, since the material is impermeable to vapor, the vapor cannot pass through as such and therefore there is no connecting vapor pressure line between *C* and *D*. Materials of this nature absorb water directly from the vapor with which they are in contact. The percentage of water which they will absorb does not depend upon the absolute vapor pressure but does depend on the per cent of saturation or, in this case, the relative humidity of the air.

From data published by the United States Forest Products Laboratory the moisture equilibrium content of wood in contact with 80 F, 50 per cent relative humidity air would be approximately 9 per cent by weight and, when in contact with air at 30 F and 70 per cent relative humidity, it would be slightly over 13 per cent by weight. Thus the line *E-F* shows the probable moisture gradient through the wall which would be established by the air conditions shown. From this it is evident that the moisture travel through this wall would be from right to left, or from the side in contact with the low vapor pressure air to that in contact with high vapor pressure air. This is contrary to the usual assumption that moisture travels through a wall in direct proportion to the vapor pressure drop between the two sides of the wall. It should be remembered, however, that in this case the vapor does not pass through the wall but that it is condensed and absorbed by one surface of the wall, transferred through as free moisture, and evaporated from the other surface. The direc-

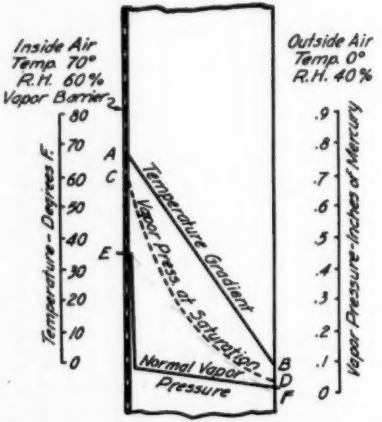

FIG. 4. WALL BUILT OF NON-HYGROSCOPIC, HOMOGENEOUS MATERIAL PERMEABLE TO WATER VAPOR. VAPOR BARRIER PLACED ON HIGH VAPOR PRESSURE SIDE OF WALL. NO CONDENSATION WILL OCCUR

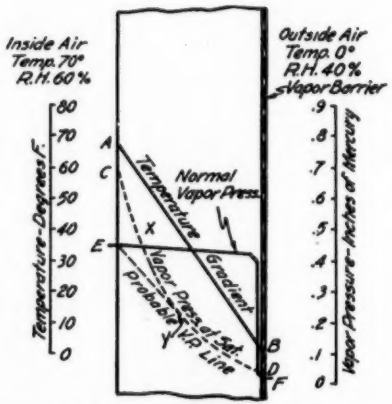

FIG. 5. WALL BUILT OF NON-HYGROSCOPIC HOMOGENEOUS MATERIAL PERMEABLE TO WATER VAPOR. VAPOR BARRIER PLACED ON LOW VAPOR PRESSURE SIDE OF WALL. CONDENSATION WILL OCCUR BETWEEN POINTS *Y* AND *F* THROUGH WALL

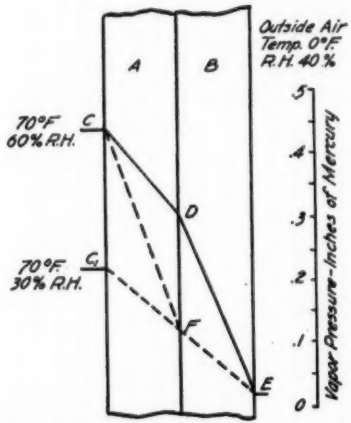

FIG. 6. WALL BUILT OF TWO MATERIALS EACH NON-HYGROSCOPIC AND PERMEABLE TO WATER VAPOR. VAPOR RESISTANCE OF *B* EQUALS TWO TIMES THE VAPOR RESISTANCE OF MATERIAL *A*

tion of vapor travel is determined by the moisture equilibrium conditions for the two sides of the material and not by the absolute vapor pressure differences. Wood is a very common example of a hygroscopic material used in buildings which has a very low permeability to vapor. In most cases the sheets of wood are not continuous and vapor may travel through the openings by diffusion or convection.

In most practical cases where the vapor pressures are different on the two sides of a building wall a vapor pressure gradient will be established through the wall, and the vapor will travel in the direction of vapor pressure drop. The slope of the vapor pressure line for different sections of a wall will depend upon the resistance to vapor passage at various parts of the wall. Figs. 4 and 5 represent two walls which are built of homogeneous non-hygroscopic materials which are permeable to water vapor. Both are subjected to air on the warm side at 70 F and 60 per cent relative humidity, and air on the cold side at 0 F and 40 per cent relative humidity. The inside or warm surface of Wall 4 is lined with a material which has a high resistance to vapor penetration, and for Wall 5 this vapor resisting material has been applied to the outer or cold surface of the wall. The normal vapor pressure gradient which would be established by the vapor pressures on the two sides of the walls is shown by the broken lines *E-F*. Since the wall of Fig. 4 has a high vapor resistance on its warm surface, the vapor pressure within the wall is comparatively low, but in Wall 5 with the vapor resistance on the cold surface the vapor pressure within the wall is high. If the temperatures throughout both walls were at all points above the dew-point temperatures of the vapor at these points in the wall there would be no condensation. For the temperature conditions shown, however, the high vapor pressures within Wall 5 are above the maximum allowable vapor pressures throughout the greater part of the wall and moisture will be formed. In Wall 4 the vapor pressures are below the danger point throughout the wall and no vapor will condense. From the discussion of the condition shown in Fig. 2 it is evident that the line *E-F* of Fig. 5 does not represent the true vapor pressure gradient established in the wall. The actual line cannot be above the curved line *C-D*.

Fig. 6 represents a wall built of two materials, *A* and *B*, both of which are permeable to water vapor. If the vapor resistance of *B* is twice that of *A*, and if the horizontal lines *C* and *E* represent to scale the vapor pressure on the corresponding sides of the wall, then the broken line, *C-D-E*, represents the normal vapor pressure gradient that would be established through the wall due to the vapor pressures on the two sides. If the temperature at the central section of the wall, *X-Y*, is high enough to allow the vapor pressure, *D*, without condensation, then the vapor will be transmitted at the same rate through all parts of the wall and there will be no condensation within the wall. If the temperature at the central section of the wall is lowered until the maximum vapor pressure that can be carried is at some lower point, *F*, then the vapor pressure line, *C-F-E*, will be established and evidently the rate of vapor travel through material *A* will be increased and that through material *B* will be decreased and there will be an accumulation of moisture within the wall.

If the vapor pressure on the left side of the wall is decreased to a point C_1, which is 50 per cent of that at *C*, and which will result from lowering the

relative humidity of the air in contact with this surface to 30 per cent without changing its temperature, then the pressure gradient through material *A* would be *C'F* and the amount of moisture transferred to the central plane of the wall would be decreased, but since the vapor pressure at *F* is still determined by the same wall temperature as previously the vapor pressure gradient *F-E* is not changed and therefore the rate of vapor travel from the central section to the outer part of the wall is not changed. It is evident that there will be a reduction in the rate of accumulation within the wall and the percentage of this reduction will be greater than the percentage by which the vapor pressure on the left side of the wall was reduced. Thus, when condensation occurs within

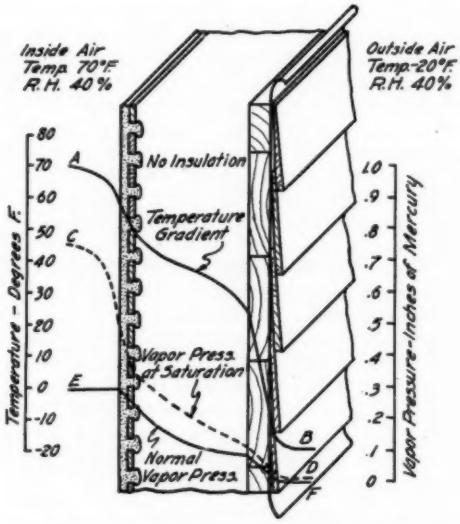

FIG. 7. FRAME WALL WITHOUT INSULATION SHOWING NORMAL TEMPERATURE AND VAPOR PRESSURE GRADIENT THROUGH WALL

a wall, a reduction in the relative humidity on the warm side of the wall will reduce the rate of condensation by a greater percentage than the percentage reduction in relative humidity providing that there is no change in temperature.

Figs. 7 and 8 represent two typical frame walls which are identical with the exception that Wall 8 has been insulated with a fill insulation between the studs. In both cases the air on the warm side is at 70 F and 40 per cent relative humidity, and that on the cold side is −20 F and 40 per cent relative humidity. The temperature gradients through the walls are shown to scale by the lines *A-B;* the maximum vapor pressures that may be carried at any point within the wall for the corresponding temperatures are shown by the dashed lines *C-D;* and the probable vapor pressure gradients established by the vapor pressure differences on the two sides of the wall are shown by the lines *E-F.*

The lines *E-F* are shown as curved lines through the central section of the walls as it is probable that the vapor is carried through this section partially by convection and not entirely by molecular diffusion.

The addition of insulation to the wall of Fig. 8 reduced the temperature of the inside surface of the sheathing from approximately 26 F to −6 F. This reduced the maximum vapor pressure which could be carried at this point without condensation. In the wall of Fig. 7 the vapor pressure line *E-F* does not cross the limiting pressure line *C-D*. In the wall of Fig. 8, however, the surface temperature of the sheathing has been reduced to such an extent that

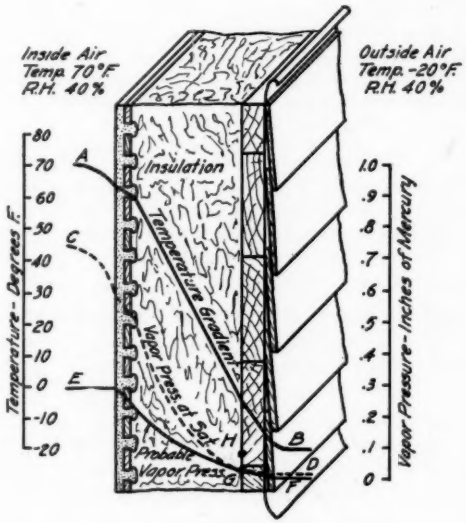

FIG. 8. FRAME WALL WITH INSULATION SHOWING TEMPERATURE AND VAPOR PRESSURE GRADIENTS AS ESTABLISHED THROUGH WALL. *H* INDICATES VAPOR PRESSURE REQUIRED AT INNER SURFACE OF SHEATHING TO PREVENT CONDENSATION. *G* INDICATES MAXIMUM VAPOR PRESSURE ALLOWED BY TEMPERATURE OF SHEATHING

the maximum vapor pressure that can be carried on the inside surface of sheathing shown by *G* is less than the vapor pressure *H* required for an equal rate of vapor travel through all sections of the wall. This means that the rate of vapor travel from the inside or warm surface of the wall to the sheathing will be increased as the vapor head, *E-G*, is greater than the vapor pressure head, *E-H*, and that the rate of vapor travel from the surface of sheathing to the outside or cold surface of the wall will be decreased as the vapor head, *G-F*, is less than the normal required vapor head *H-F*.

When vapor accumulates at any point within a wall it is due usually to the fact that the vapor travels through the warm section of the wall to the accumu-

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lation area at a greater rate than it travels from this area to the cold side of the wall. If it is assumed that the temperatures are fixed, then the rate of condensation may be reduced first by reducing the rate at which vapor travels from the warm side of the wall to the area of condensation, or second by increasing the rate at which it may travel from this area to the cold side of the wall. The rate at which vapor travels from the warm area to the inner section of a wall may be reduced first by reducing the relative humidity and thus the vapor pressure on the warm side of the wall, and second by introducing some material with high vapor resistance on the warm surface of the wall. The rate at which vapor may travel through the outer section of a wall may be increased by reducing the relative humidity on the outside of the wall, (this is usually impractical or impossible), or by constructing a wall with outer surface materials which have low resistance to the passage of vapor. One of the most effective methods for reducing the moisture accumulation within a wall of a building is to reduce the relative humidity of the air in contact with the warm surface of the wall.

In the past it has been common practice to use a building paper or some similar material on the outside surface of a wall to protect it from free moisture in contact with the outside surface and to reduce air leakage through the wall. Recently many of the building papers have been improved and are now good vapor barriers, thus preventing the escape of moisture which may pass through the inside surface into the wall.

All of the laws governing the flow of vapor through materials have not been demonstrated by experiments, but data available indicate that in many cases the flow of vapor through a material or combination of materials is proportional to the vapor pressure drop along the path of flow.

From the nature of certain materials and from the fact that vapor may be condensed either by lowering the temperature or by the hygroscopic property of many materials, it is evident that there will be many different conditions under which the flow of vapor may not follow any simple law.

DISCUSSION

E. C. LLOYD (WRITTEN): The author has discussed a subject which has many ramifications and covers many points that are of interest to all concerned in insulation problems. Today that means that the subject matter is of real interest to almost everyone whether he is actively engaged in insulation work or whether he is confronted with some condensation problem. I find that in this age of refrigeration and air conditioning the condensation problem affects all of us in some way or another.

Professor Rowley's theory is nicely stated and well developed, but before this problem can be completely solved by the graphic method proposed and expounded, it will be necessary to explore further the field of possibilities. To me there are several points which might well be considered further at this time. First, in the examples shown in Figs. 4 and 5 the important factor to me is not so much the efficacy of the vapor barrier in terms of the efficiency of the paper or other membrane employed as vapor seal as it is the over-all effectiveness of the installation of that seal on the structure. This over-all effectiveness introduces the second point which is the effect of actual air movement in a wall on the temperature

gradient through that wall, and also the effect of the same air movement on moisture passage. In short, we may have a structure which reverses the usual practice and carries a low temperature and low humidity inside while the exposure is to summer heat and high humidities outside. In such a structure an attempt to seal on the cold side with conventional building papers is defeated by the weakness of the joint between sheets of paper combined with actual air pressures produced by the four winds. The air passage carries moisture into the wall at some points and out of it at others. Prevailing winds from one general direction invariably result in a higher proportion of moisture in one side of the building than in the other, so that it is my opinion, based on practical experience, that the best way to install insulation on a structure where the service conditions are unusually severe is to provide a solid wall and seal the insulation not only on the warm side, but on the cold side as well. My third point is one that warrants some further consideration. I refer to the effect of absorbed water on the thermal conductivity of the so-called hygroscopic material and the effect of the increased heat flow coming as a result of the absorbed moisture on the temperature gradient. Moisture in the building wall normally accumulates and as it builds up in moisture content, the wall becomes a better conductor, hence, the temperature gradient changes. That this moisture take-up and the consequent changes in transmission as well as the increase in moisture accumulation are important, goes without saying.

It is admitted that the points raised add complications to an already complicated subject. Again let me say that the author has made an excellent start, but care should be used in accepting all of the information given without due consideration of other points such as those raised here.

Another type of structure which should be given consideration is the one where the warm and cold sides of the insulation reverse with the seasons. This is true of many industrial structures and also in the well insulated, completely air conditioned residential structure. Again for such cases it is my idea that only a thorough sealing on both sides will serve to hold the insulation intact and provide the necessary protection to the structure.

Without question the paper adds a great deal to the solving of the problems presented by condensation in insulation, and what we should have now is the actual application of these theories by examination of performance of various types of structures under highly humid operating conditions.

W. V. HUKILL[1] (WRITTEN): The author's use of the terms *permeable* and *permeability* seems to be unusual. Can any materials impermeable to water vapor absorb or transmit moisture? Its permeability to air or other gas seems to be beside the point, since, as the author points out early in his discussion, water vapor in the air has the same physical properties that it would have if the air were not present.

In the example shown in Fig. 3, line E-F shows a moisture content gradient and does not necessarily indicate the direction in which moisture is moving. Is it not likely that moisture is moving from the left to the right and, therefore, in accordance with the vapor pressure difference between the two sides, in spite of the direction of slope of the moisture content lines? Is it not also reasonable to suppose that at every point in the wall a definite vapor pressure exists and that a valid vapor pressure gradient (line *C-D*) can be pictured?

When the vapor pressure of a hygroscopic material is defined as equal to the vapor pressure of an atmosphere with which it would be in equilibrium if the temperatures of the material and the atmosphere are equal, then the vapor pressure of any part of the wall pictured in Fig. 3 has a definite meaning. The vapor pressure of wood, for example, can be calculated from data such as that quoted on page 549, being a function of moisture content and temperature. Assuming that a vapor

[1] Bureau of Agricultural Chemistry and Engineering, Washington, D. C.

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pressure gradient does exist through the wall, the distinction between moisture flow resulting from permeability (as the term is used in this discussion) and flow by capillarity or some similar process is not essential. The flow in either case should be expected to be from high vapor pressure to low vapor pressure.

From the statement regarding the line *E-F*, Fig. 3, it is evident that the moisture travel through the wall would be from right to left, or from the side in contact with the low vapor pressure air that in contact with high vapor pressure air is apparently based on the reasoning that moisture must flow from higher moisture to lower moisture content. Without further support for this reasoning, the reverse direction of flow is not evident.

The author does not state that actual flow measurements have been made under the conditions illustrated in Fig. 3. In the absence of experimental demonstration the assumption that moisture flow is in the direction of a downward moisture content gradient instead of in the direction of a downward vapor pressure gradient does not seem justified. I should be glad to know if any tests have been made to demonstrate this apparent reversal. If they have, and if they bear out the author's conclusions, I believe a future paper giving a fuller discussion of the theory of such flow and of the laws by which the direction of moisture travel is evident, as well as an explanation of the somewhat confusing conception of a material which is impermeable to water vapor, yet which transmits moisture, would be very helpful.

R. H. HEILMAN (WRITTEN): This paper is a valuable contribution to the literature, contributing considerably to the more complete understanding of vapor transfer and its effect upon the condensation of moisture within building walls.

The following brief discussion is not a criticism of this paper, as the main purpose of the paper is vapor transfer and not temperature gradients. However, an examination of Figs. 7 and 8 indicates a difference of only about 5 F for the interior wall temperatures of the insulated and uninsulated walls. Actually the difference should be about 11 F. If the theory of warm walls means anything, then it is believed that the higher wall temperatures resulting from the use of insualtion is of considerable importance.

Using ASHVE constants, the calculated inner wall temperatures for the insulated and uninsulated walls should be approximately 66.6 F and 55.1 F, respectively, for an interior temperature of 70 F and an exterior temperature of −20 F.

In a paper[2] in 1934 Professor Rowley showed that increased radiation of heat from occupants in a room to cold walls required a higher air temperature for the same feeling of warmth than was required for a room with warm walls.

P. F. MCDERMOTT[3] (WRITTEN): We wish to express our appreciation of Prof. Rowley's fine work in preparing this paper which is particularly interesting and valuable because of the number of half formed theories prevalent in this confused field. We feel that this paper will do much to bring orderly thought to this subject, and to produce well crystallized theories.

In the main, we agree with the theory presented by Professor Rowley. We hope in this discussion to express some further details of our theory rather than to criticize those presented.

We question the hypothesis that gas transfer through non-absorbing permeable materials, as induced either by small partial pressure differentials or small total pressure differentials, will be found of such nature as to permit assigning resistances which may be used as are resistances to heat transfer. Fundamentally the hypothesis

[2] ASHVE RESEARCH REPORT No. 996—Insulating Value of Bright Metallic Surfaces, by F. B. Rowley. (ASHVE TRANSACTIONS, Vol. 40, 1934, p. 413.)
[3] Physicist, Johns-Manville Research Laboratories, Manville, N. J

is to the effect that under a given pressure differential a given barrier should allow twice the transfer permitted by two such barriers in series. More simply, resistance to transfer should be directly proportional to thickness. We have conducted tests in an endeavor to determine such resistances in the hope that they might be used additively but our results were entirely negative. We know of other tests of similar indication.

We believe, in view of these tests, that the resistance of complete structures will need to be determined. However, the hypothesis should not be abandoned, for a mass of data may yield relationships which will permit reliable calculation of combination of barriers.

Professor Rowley speaks of hygroscopic materials as condensing water vapor with the intimation that such vapor becomes actual liquid water. As we understand the phenomenon of hygroscopicity this would not be strictly so. We submit here, therefore, our theories regarding this phenomenon.

In the first place, we distinguish between absorption and adsorption in that the first pertains to a *macroscopic* and the latter to a *submicroscopic* accumulation and retention of the liquid. Absorbed water results in a sensible wetness of the material whereas adsorbed water would not. An example of these two is had in a sponge which upon immersion in water both absorbs and adsorbs. Upon squeezing the sponge the greatest part of the *absorbed* water is forced out but the sponge retains an appreciable amount of water which cannot be ejected. This residual water is the *adsorbed* water. The absorbed water is that which fills the openings within the material and which does so largely through cohesive forces between water molecules. The adsorbed water is that which covers the surfaces of the material in a layer of the order of one molecule in thickness and which is retained by adhesion between molecules of water and of the material.

These so called adhesive forces are frequently considerably greater than the forces existing between water molecules and evidence has been reported indicating that the former, in some cases, attract water molecules from the vapor to such an extent as to create a surface layer and in addition an adjoining region of considerably more dense vapor.

This attraction is generally considered to be of a nature somewhere between gravitational and chemical attraction. It is not either of these in the pure sense nor yet is it solution of the adsorbed in the adsorber nor simply a surface tension phenomenon. We do not know exactly what to consider it and are not offering any further theory as to what it may be.

The point is, however, that this not too well defined mechanism bears on the migration of water molecules through materials and further that migration by it may be opposite in direction to that induced by vapor pressure differentials.

To consider a specific instance to explain this theory let us think of a fiber or group of fibers extending from a region A to a region B which are regions of high and low relative humidities respectively. We will find a transfer of moisture to take place from A to B. Now if region A is sufficiently cooler than B the vapor pressure may be greater at B than at A and the flow is contrary to the vapor pressure differential. This requires then that the fibers act to extract vapor at A and release it at B. Thus, at A some force is necessary to pull molecules from the low pressure vapor, to transmit them along the fiber and force them into the vapor of somewhat higher pressure at B.

This can be, and sometimes is, accomplished by hydration and dehydration of the fibers at their ends at A and B respectively, and we believe such cases are readily understandable. However, since we have the transfer in the absence of actual hydration other causes must be found.

We propose that the motivation is one of adsorption in which adhesion supplies the forces necessary; that this greater affinity serves to extract water molecules

from the region of greatest relative abundance or greatest relative humidity pass them along the fiber and release them to the region of least relative abundance or least relative humidity.

The layer of molecules about the fibers need never be condensed into liquid to accomplish this. In fact, if liquid were postulated it would be more difficult to understand the transfer since a liquid film would be thought of as sufficiently thick that its surface layer would exhibit the saturation vapor pressure. In which case, it could not effect the extraction of water from an unsaturated vapor.

We have considered fibers in this example for simplicity and it is clear that all we require are interconnecting surfaces between our two regions. In fact, occasional discontinuities can be included in our problem without disturbing the explanation.

There may be a tendency to question the pertinence of such theorizing, and in event any of you do question it just study out why cold water and other cool pipes, when covered with reasonably porous insulation, do not become entirely wet during the first few warm humid days and remain so throughout the entire summer.

This is another theory which may not be correct, but this one and Professor Rowley's will serve as a basis of thought until one or both are proved incorrect.

We would appreciate Professor Rowley's detailed explanation of Fig. 2 in which, while describing the conditions within a wall where saturation interferes with what would otherwise be the normal vapor pressure gradient, he constructs a tangent to a curve and has it pass through a point not on the curve. This in itself is mathematically incorrect; however, our divergence of opinion is with the tangent idea. We do not believe the conditions require anything more than a point tangent and in fact due to the generally low resistance to transfer across these interior spaces we believe that the vapor pressure throughout them is more likely to be essentially that fixed by the colder surface.

In another instance reference is made to moisture within a material as free moisture. We feel that *free* moisture would act as liquid or vapor rather than as moisture bound by adsorptive forces to the material and, therefore, question the use of the term *free*. In other words the action described would not take place if the moisture were *free* and not influenced by other forces.

Professor Rowley seems entirely justified in his analogy between vapor and heat transfer across air spaces. As he points out (except the heat transfer by radiation) both are accomplished by molecular diffusion on which is imposed convection. Dr. G. A. Downsbrough of our Laboratory has calculated that the net transfer across a space 4 in. wide, having a constant water vapor pressure differential of 10 mm of mercury, would amount to approximately 2,000 gm (grams) per square foot per day by pure diffusion. Vapor barriers are required to pass not more than 0.5 gm per day under these conditions thus we see that diffusion alone constitutes a good enough carrier as to eliminate an air space as a resistance.

We have wondered for a long time what the velocity of this diffusion would be, with the thought that were convection replaced by a flow of the total mixture across the space from low to high vapor pressure sides, some minimum velocity would be had above which up stream diffusion would be impossible. In other words, we are convinced that a total flow through a wall, from low to high vapor pressure sides, could be of sufficient velocity to prevent the transfer by diffusion. This might be of value if the velocity were low enough to make such a preventative method practical. So far, however, attempts to arrive at the velocity of diffusion have failed.

WHARTON CLAY (WRITTEN): This paper gives theoretical explanation to the experiences which are observed by mineral wool manufacturers in the action of exterior walls filled with mineral wool exposed to excess humidities, and without

benefit of a moisture vapor barrier. The author, as applied to this specific problem, explains why mineral wool does not absorb moisture from the atmosphere.

Mineral wool is a non-hygroscopic material and will not absorb water vapor directly from the air. Thus any condensation which may be caused by the application of mineral wool to the wall is due to the fact that the heat leakage has been stopped and not to any hygroscopic property of the insulation.

Experience has shown that in severe cases where condensation does occur it takes place on the sheathing or other surface finish of a wall rather than in the insulation even though calculations would show that the normal vapor pressure gradient through the wall would cross the maximum allowable vapor pressure line at some place within the insulation. The theory presented in this paper explains this action.

The only way that mineral wool or other non-hygroscopic material could become wet and cause the uninformed to claim that the moisture had been absorbed from the atmosphere is by the condensation of water vapor on some cold surface, such as the inside surface of sheathing of an ordinary frame wall from which the water may be absorbed. If this condition has been brought about by the application of insulation it is due entirely to the fact that heat has been saved and not to any quality of the insulation. The remedy is not to condemn the insulation but rather to keep the vapor out of the low temperature area. This can be done by effective vapor barriers on the warm side, or to a certain extent by ventilation to the exterior of the wall.

The entire situation, however, is unnecessary if humidities are kept below the following recommended limits:

OUTSIDE TEMPERATURE	INSIDE RELATIVE HUMIDITY
Below 0	Not more than 20 per cent
0 and 20 above	Not more than 30 per cent
20 above and over	Not more than 40 per cent

Much has been said about the problem; however, it is rare to find condensation troubles even among 400,000 installations made before the vapor barrier principle was recognized.

W. A. DANIELSON (WRITTEN): Professor Rowley presents *a* theory but there is every reason to believe that he has *the* theory which agrees with physical laws. It is now possible for a designer to definitely plan for proper insulation where vapor pressures differ on the two sides, which is almost invariably the case.

The rate of transfer of water vapor through a wall is dependent on the character of the material, its thickness and the vapor pressure between the two sides. Where there is an impermeable barrier then there is no transfer of water vapor through the wall but there is a transfer into the wall, and if at any point in the wall the dew-point is reached then there will be condensation.

The writer is able to more readily analyze the conditions by plotting the dew-point temperature. For instance on Fig. 6 if the line *D-F* is the vapor barrier and the dew-point temperature corresponds to 70 F and 60 F relative humidity is plotted, it can be determined readily whether or not there will be condensation on the warm side of the vapor barrier. Should the dew-point line strike above *D* there will be condensation; if below *D* there will not be. If the insulation of *A* is increased, a point would be reached where the dew-point temperature would be above *D* and there would be condensation. It is therefore evident that there is a relationship between the total insulation of *A* and *B* that can be maintained so that no condensation will occur, and should condensation occur the insulation value of *A* will be reduced and the temperature at *D* will change so that at some point on the line

C-D the dew-point temperature will cross and this will be the limit of condensation. If there is capillarity in the material the condensed moisture will of course reduce further the insulation of *A*.

The writer has just placed a vapor barrier made of coal tar and felt, similar to a built-up roof, in a rather large cold storage plant. On top of this was placed a layer of approximately 8 in. of sawdust. Previous to this the ceiling below, which is just above the cooling coils, was always saturated to the extent that icicles quite frequently formed. Since the barrier has been placed the wood shows every evidence of drying out, there is no moisture other than frost and this frost is evidently due to the loss of radiant heat to the cooling coils, so that this surface is actually cooler than the air and the frost is the result of condensation from the air within the box. Since this change was made the operator states that he is able to maintain a low temperature easier than previously.

C. F. MALLY: Professor Rowley points out that the passage of vapor through walls is affected by materials which offer resistance to this passage.

While his recommendation is the application of vapor barrier materials on the inside the walls, it seems the general practice of asphalted fibre sheathing boards applied on the outside walls should be criticized, in view of the vapor which will condense on the surface of this material in extreme temperatures.

While Professor Rowley has proved this theory which is generally accepted, proper steps should be taken to use these valuable facts to improve home construction.

PROFESSOR ROWLEY: As pointed out in this discussion there are many variables entering into the theory covering the transfer of vapor through materials. The mechanisms by which vapor is transferred may become very complicated and when the theory is applied to practice it will often be necessary to consider the problem from a practical viewpoint rather than entirely from the theoretical point of view. It is easier to find a satisfactory, practical solution of any problem when the theory is understood and taken into consideration than when it is not. The purpose of this paper has been to bring out a discussion and try to clear up some of the theory which has been advanced covering the condensation problem. More experimental work will be required to either prove or disprove the theory. However, insofar as the research work done at Minnesota comes within the scope of the theory discussed, the results have substantiated the theory. The difficulty of selecting materials which have ideal properties for proving or disproving the various theories and of accurately measuring the rate of vapor flow appears to be even greater than those which were originally experienced in developing the theory and practice of heat flow through material. The discussions and criticisms from the various authors are appreciated.

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FIRE PROTECTION FOR AIR CONDITIONING SYSTEMS

By R. C. Loughead,* Detroit, Mich.

IT is presumed that members of the society have all the knowledge and facilities necessary to install an air conditioning system so that it will function properly and efficiently and provide clean, fresh air at the proper temperature and humidity; and that the fire prevention engineer knows little about the engineering problems of the air conditioning profession. It is known, however, that with proper design and planning, the potential fire hazard associated with air conditioning may be minimized and to a large extent eliminated and fires entailing loss of life may be prevented.

The regulations of the *National Board of Fire Underwriters* (pamphlet No. 90 with revisions of May, 1939), cover the installation of air conditioning, warm air heating, air cooling and ventilating systems, and these regulations provide for the mitigation of the fire hazards in connection with air conditioning. It is the purpose of this paper to endeavor to point out the necessity of regulation based on fires which have already occurred due to faulty design and installation and to provide heating, ventilating and air conditioning engineers with sufficient information relative to the inherent fire hazards which exist with respect to air conditioning so that systems may be more generally properly designed and installed from a fire prevention standpoint.

A system installed in a large department store provides conditioned air to the sub-basement, basement, first floor, and mezzanine, with the conditioning equipment located in the sub-basement. Such an installation, regardless of its design, might include more potential fire hazards than any other type since it serves more than one floor from below so that gravity, in conjunction with the direction of air flow, tends to increase the possibility of combustible materials entering the recirculating ducts. But this particular system is installed according to the regulations of the *National Board of Fire Underwriters* and is therefore presumed to be properly protected. The ducts are constructed of incombustible material, they are accessible for inspection and cleaning, protected where in the vicinity of combustibles, lined and lagged in part with incombustible materials, and provided with automatic fire dampers at floors, fire walls, and partitions.

* Chief Engr., Michigan Inspection Bureau.

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The recirculating duct openings are covered by screen of small mesh so that it is impossible for other than dust or lint to enter these ducts. The filters are incombustible, kept clean, and protected by a sprinkler system actuated by a manually reset thermostat which is also connected to automatic dampers, cutting off the plenum chamber and stopping the fan. Steam coils for heating are properly protected, and brine is utilized for cooling. This is a standard system, installed according to the regulations of the *National Board of Fire Underwriters,* and there have been no fires in connection with an air conditioning system installed according to these standards.

Generally speaking, the system described is a paradox, a "pipe dream," for such a system with the protection described, is practically non-existent, although these essential installation features are necessary if life and property are to be protected from fire in air conditioned buildings. That such a protected system does not exist to a greater degree is not the fault of engineers who design and install but rather the responsibility of the fire prevention engineer who has failed to educate them to design with proper fire prevention and pro-

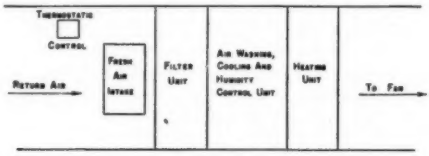

FIG. 1. RELATIVE LOCATION OF APPURTENANCES

tection features. This safe design will more generally follow when the lessons learned from fires which have occurred are thoroughly understood, and this can be accomplished at a relatively small increase in cost. But before discussing fires which will serve as illustrations of improper design or faulty maintenance, it might be well to point out the different parts of an air conditioning system in which potential fire hazards may exist.

A typical air conditioning system may consist of the following parts: a fan producing the movement of air; a system of ducts located on the discharge side of the fan, used for conveying the conditioned air to the desired locations; a return system of ducts from these locations back to the conditioning units; a fresh air intake for supplying additional air from the outside; a series of filters in the return and fresh air ducts for cleaning the air; a water spray for washing, humidifying, and cooling the air; steam coils for raising the temperature; refrigeration equipment for cooling the water spray or for cooling the air directly or indirectly by means of coils in the ducts; and the valves, controls, and other appurtenances for the automatic operation of the system. Fig. 1 shows a schematic diagram of the relative arrangement of the component parts.

The fan or blower should be so located and arranged that ready access is afforded for repairing, cleaning, lubrication, and inspection. One or more controls, readily accessible, should be provided so that the fan may be promptly shut down in case of fire and an automatic fire detecting device should also perform this function. Due to the relatively high air velocity in the ducts,

sometimes as high as 1200 fpm, materials which become ignited burn more vigorously on account of the excess oxygen.

The discharge ducts which convey the treated air to the areas conditioned may consist of many feet of ducts of various sizes, terminating in meshed or grilled openings, usually located at a considerable distance above the floor, which, in combination with the direction of air flow, generally precludes the possibility of bits of paper or other combustibles entering these openings. However, more or less dust may collect therein, depending upon the efficiency of the filters. Certain types of filters which permit the passage through them of oil soaked lint should not be used since there is a possibility of combustible material lodging in these ducts, and in any event means for periodic inspection and cleaning should be provided. In general, the inherent hazard of the discharge ducts is relatively small, but they may be the means of carrying fire, hot gases, and refrigerants into the conditioned areas from other points and therefore may require protection.

The return or recirculating duct system is usually as extensive as the discharge ducts and, if improperly designed and installed, may present a fire hazard much greater than any other part of the system. When the openings from the conditioned areas are at or near the floor, this, together with the direction of the movement of air, may invite the tossing of burning cigarettes, bits of paper, and other combustible debris through the grille openings; therefore, the return ducts should be provided with inspection and clean-out openings so that they may be kept clean. The mesh of the screen or grilles should be made as small as possible without reducing the efficiency of the system, and in new designs this may be accomplished by increasing the duct inlet areas. Recognition should always be given to the added possibility of a fire originating anywhere in the conditioned area being drawn through these ducts, involving any combustibles within the system and then being conveyed into all parts of the building.

Each conditioning system is usually provided with a fresh air inlet, which should be of fire resistive construction, used for no other purpose, and the opening to the outside should be located well away from any possible exposure or alleys and areaways where burning materials or smoke may be taken into this duct and distributed throughout the system.

In general, when a duct system is installed at the time the building is erected, the ducts may be concealed in the structure, but otherwise the vertical ducts are usually installed in the building columns and the horizontal ducts suspended from the ceiling. In any event, all ducts should be constructed of incombustible materials and well separated from woodwork and other combustibles. When lining or lagging is used for the minimizing of heat losses or the reduction of noise, this lining or lagging should be of incombustible material to eliminate the fire hazard due to combustible linings. Even though the materials used may not be highly combustible under ordinary conditions, the possibility should be recognized that they may burn vigorously in the presence of the air movement in the ducts.

There are a number of different types of filters, only a few of which are entirely incombustible, but in any event their function is to collect dust and

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lint which are combustible. Some filters are made up of units of steel wool, spun glass, rock wool, or a series of screens or metal surfaces set into frames and in order that they may be efficient they may be dipped or sprayed with a high flash oil whose function it is to trap the dust particles. Another type consists of a revolving screen which passes through a reservoir of oil. The air stream type consists of a series of staggered rods covered with oil, the oil picking up the dust. This type may present an added hazard, allowing an unusual amount of oil soaked dust and lint to pass through the filter and be deposited in the discharge ducts. All of these types are more or less hazardous, even when clean, on account of the oil. Another type is built in units consisting of cloth and cotton or cellulose so arranged in a zig-zag manner as to present a large surface to the air, and while no oil is required in connection with these filters, they are combustible and a source of fuel should a fire reach them. A cloth of spun glass, which is incombustible, is now available for this type of filter and is probably the least hazardous of any filter on the market at the present time. The ultimate aim of the filter manufacturer is to discover a liquid which will not burn and at the same time possess all the filtering qualities of oil.

Steam coils located in the main duct for heating the air are usually of the fin type and should be located well away from combustible materials.

Refrigeration in connection with air conditioning may be used for cooling water sprays, for cooling brine which is circulated in coils in the ducts, or the refrigerant may be expanded directly through these coils, in which case the failure of the coils releases the gas into the duct system and in turn to the conditioned areas. Many of these refrigerant coils are made up with solder which generally has a relatively high melting point but nevertheless would fail under actual fire conditions, releasing the gas which may be explosive or toxic.

The purpose of the description given is to show that there may be a decided fire prevention problem in connection with air conditioning and to recommend the regulations of the *National Board of Fire Underwriters* for the protection of these hazards. In addition to these safeguards, constant vigilance should be practiced so that the chance of a fire involving the entire system is remote. Cleanliness is of paramount importance.

For some time the Underwriters' Laboratories have listed refrigeration units for use in connection with air conditioning, but it is only relatively recently that they have developed a fire test procedure for filters and now they are listing unit filters under the following classifications:

Class I filters are those which when clean do not contribute fuel when attacked by flames and which emit only negligible quantities of smoke.

Class II filters are those which when clean burn moderately when attacked by flames or emit moderate amounts of smoke, or both. These filters may be consumed to some extent but will not project flame or sparks down the duct during the test.

This means that filters which are readily flammable or which when tested project flame or sparks downstream, or which emit more than moderate

quantities of smoke or emit products of combustion which are unusually disagreeable or toxic, are not considered as suitable for listing by the Underwriters. Quoting from the Laboratories standard, the test equipment is as follows:

The duct used in the test is of galvanized iron construction, reinforced with angle irons. It is 21 in. wide by 41 in. deep and 27 ft 10 in. in horizontal length. At one end it is tapered to connect to the discharge of a variable speed, electrically driven blower. The other end of the duct is an open discharge. At about the middle of the duct are frames to receive two 20 by 20 in. filter units. A tight fitting door permits access to the filter frames and numerous glazed openings permit observation of both sides of the filters and of conditions in the duct downstream from the filters. Four 1-in. elbows form burner outlets for the city gas fuel, so arranged that two flames play on each filter. The flames are yellow and wavering and extend intermittently about 12 in. beyond the filter frames when the latter are empty. The burners consume about 9 cu ft of gas per minute, having a Btu value of about 805 per cu ft.

RESEARCH NEEDED

Several manufacturers are developing equipment for the detection of smoke, and the *National Board of Fire Underwriters* is sponsoring tests and research to determine effective means of detecting and controlling spread of smoke in air conditioning systems and to reduce the possibilities of smoke damage and panic hazard. These devices are intended to be used to shut down blowers, close dampers, sound alarm, and perform other necessary functions to prevent the spread of fire and smoke by the air conditioning system. Such a system has been installed in three large air conditioning systems in a Detroit department store and will be supervised and regularly inspected and maintained by a central station company operating in that city.

One of the basic principles of fire prevention engineering is the division of a building into fire areas by fire walls and protection for all communicating or vertical openings so that the loss in any one fire may be minimized. It has long been realized that air conditioning which entails the piercing of these fire walls and floors with ducts would offset much of the effort which has been made to reduce the areas which may be involved in a fire. The actual regulation of theaters, motion picture studios, paint spraying, etc., came about only after there was loss of life by fire in these occupancies, and it is hoped that lessons such as these will not be necessary in order that the installation and use of air conditioning may be properly regulated. While there is no record of loss of life associated with improperly designed systems at this time, the fire loss and smoke damage has been considerable.

CARELESSNESS CAUSES FIRE

A fire occurring in a Detroit department store over a year ago, causing a loss of about $22,000, illustrates the fire hazard in connection with improper type filters. In this case, the filter involved consisted of a revolving screen passing through a reservoir of oil, which was carelessly ignited by an acetylene torch. After the fans were stopped, most of the smoke and heat, very fortunately,

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backed up through the fresh air intake shaft and away from the interior of the store, reducing the loss to an appreciable extent. This does not mean that filters of this type should be condemned, but that, on account of the large quantities of oil, they are not adapted to occupancies where the panic hazard is pronounced. Other fires show that combustible filters, incombustible filters covered with oil, or even filters entirely incombustible but loaded with dust and lint which may become ignited, may be a source of danger, and should be suitably protected. Records of still other fires illustrate the fact that while filters may be a potential fire hazard, they do not necessarily constitute the greatest fire hazard, which may be associated with other substandard features.

Two fires occurring in widely separated localities, New York and California, were caused by combustible duct lining. In the one in New York City the burning of rubbish in an areaway directly under the discharge duct of the air conditioning system of a theater caused smoke, sparks, and heat to be discharged into the balcony. Fortunately there was only a small audience when the fire started. The other fire occurred in a Los Angeles office building, and when the janitor discovered that the air conditioning system was emitting smoke due to burning duct lining, it was necessary for him to go from the basement to the roof of this 10-story office building in order to shut down the fan. Proper design and reasonable foresight would have made this unnecessary.

In March of this year a $150,000 loss occurred to a school building in Minnesota which might have been minimized or entirely eliminated with proper design of both structure and air conditioning system. This school building was three stories high, of fire resistive construction except the roof, all floors and the ceiling over the third floor being of reinforced concrete construction. The conditioning equipment was installed in the roof space between this incombustible ceiling and the combustible roof. The discharge ducts to all the rooms were connected to a main discharge duct in this roof space, but the recirculating or foul air ducts exhausted directly into the roof space, not being directly connected to the heating system. All ducts throughout the building were of incombustible construction except the horizontal ducts in the roof space, which were combustible. The fire, which originated in the roof space, not only was transmitted to the school rooms through the discharge ducts, but burning brands from the roof boards dropped through the foul air ducts into the classrooms. It is entirely possible that proper design would have prevented this fire.

Last winter in a city suburban to Detroit, unsold Christmas trees were being burned in an alley and smoke from the fire, entering the fresh air duct of a theater air conditioning system, caused a small smoke loss which might have been greater had the fan been operating or had the source of the smoke not been promptly discovered.

Fire starting in a pile of kindling wood outside a school building in Texas a number of years ago entered an air intake to the ventilating system and completely destroyed the building. At a textile plant in Holyoke, Mass., smoke and soot from a burning tar kettle in the yard were drawn through the fresh air intake of an air conditioning system into several manufacturing rooms on the third, fourth, and fifth stories of the building. Soot deposited on worsted yarn in process caused stoppage of plant operations and required several hours'

time to clean off equipment and restore the plant to operating condition. Whether these fires were due to the improper location of the fresh air intakes or the improper burning of rubbish or placing of a tar kettle is a controversial subject; the fact remains that had the conditioning systems in these particular buildings been properly designed, installed and maintained, there would have been no fire or smoke loss.

In a village suburban to Detroit, considerable smoke loss occurred in a dwelling due to a forced air heating system. This new home was not as yet occupied and a sanding machine had been placed on the return duct inlet located in the floor of the living room. A fire in the sander resulted in smoke being conducted by means of the discharge ducts into every room in the building in sufficient quantities so that it was necessary to completely redecorate although the flames themselves did little damage.

Undoubtedly the fire which has caused the most comment was the one occurring in January of last year in the air conditioning system of a large Detroit department store. The system was located in the sub-basement and conditioned this area, the basement, the first floor, and the mezzanine. On the evening of the fire a number of employees were taking inventory and at the time of the fire, which occurred at 10:15 p.m., the blower fans had been shut down for over an hour. Due to an unknown cause, a fire originating in the duct system caused large quantities of smoke and some flame to be discharged into the basement and sub-basement through both outlet and inlet ducts. A total of fifteen sprinklers opened, controlling the fire. The damage, which was almost entirely due to smoke, amounted to about $110,000, and the fire department is reported to have taken several employees down ladders from the second story windows. This fire illustrates very graphically the necessity for improved installations to eliminate the possibility of a pronounced panic hazard as well as heavy contents loss in connection with this type of occupancy. It is not difficult to picture a catastrophe if a sale had been in progress during the normal operation of the system.

This fire was evidently fed by an accumulation of waste paper scraps, lint, and other refuse in both the discharge and return ducts and might have been ignited by a cigarette being carelessly thrown into a return duct or the plenum chamber. In any event, this combustible material could not have accumulated in the duct system with properly designed intake grilles and proper air filters. Only a source of ignition was necessary to start a fire and could have been prevented had the duct system been kept clean. An examination of parts of the horizontal ducts in the discharge system and not involved in the fire showed an accumulation of oil soaked lint and dust covering as much as one-third of the duct floor to the extent of ¼ to ½ in. deep. This lint and dust had passed through the filter, and sales slips, gum wrappers, cigarette butts, and other combustible materials were found in the horizontal return ducts.

It can be seen from these descriptions of fires and a study of the present regulations that the fire hazard in connection with air conditioning and ventilating systems may be great or small, depending upon the safeguards provided and the inherent hazards due to design. The system supplying two or more floors from below should receive special attention, while the system supplying

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one floor or part of a floor with all equipment on that floor might require little or no protection. A simple way by which the hazard of spread of fire and smoke through a building may be effectively reduced is to install one or more small systems on each floor in place of large systems serving several floors. In this way passage of ducts through floors is eliminated, less space is taken up by ducts, and close control of atmospheric conditions in the various sections of the building can be more easily obtained. In existing buildings this method may sometimes result in less expensive installation as well as better control of fire and smoke hazards and should have the serious consideration of air conditioning engineers.

The proper installation and maintenance of an air conditioning system should include most, if not all, of the following features:

1. Fan control conveniently located.
2. Incombustible ducts, linings, and laggings.
3. Adequate cleanout and inspection openings in duct system.
4. Small mesh screen or grille in return duct openings.
5. Fresh air intake well away from exposures.
6. Incombustible filters changed or cleaned periodically.
7. An adequate refrigeration code.
8. Fire dampers to cut off parts of system.
9. Proper fire protection.
10. Careful and periodic inspection.
11. Cleanliness and good housekeeping.

The regulations require fire doors and dampers in the ducts under certain conditions. Fig. 3 shows a fire door installation in a duct and is required where the duct passes through an unpierced fire wall or a fire wall with openings protected by fire doors, while Fig. 2 shows a damper installation, required where floors are required to be separated. When discussing this requirement with members of the air conditioning industry, it was surprising to find that the general attitude seemed to be that this was an insurmountable problem which could not be complied with, until actual illustrations were presented and it was shown that the many installations might not require such cut-offs. Probably the simplest way in which this part of the regulations might be stated is that if under local construction requirements, openings are permitted through a partition, wall, or floor without protection at such openings, an unprotected duct opening would be permitted through the same partition, wall, or floor. On the other hand, if the local building code requires with respect to a certain type of construction and occupancy that an elevator or stairway be enclosed in a masonry shaft with all openings protected by fire doors, it naturally follows that a vertical air conditioning duct passing through the floor should be enclosed in masonry with a damper in the duct outlet at each floor. Fig. 4 shows such a typical installation.

The question of the extinguishment of a fire at the filters in an air conditioning system can readily be taken care of by automatic sprinklers or some other form of fire extinguishment, but if the system covers a large area and combustible material is permitted to collect in the horizontal ducts, it is impracticable, if not impossible, to protect the hazard. About the only solution to this problem is the periodic inspection and cleaning of the duct system at

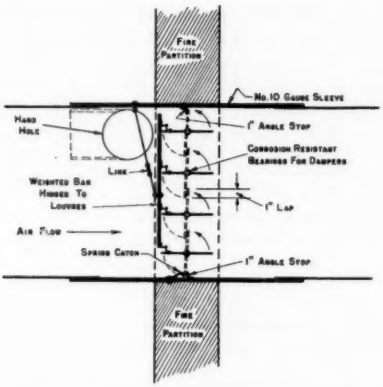

FIG. 2. DAMPER ARRANGEMENT IN FIRE PARTITION

intervals of sufficient frequency so that should a fire occur, no combustible debris will be involved.

The present regulations of the *National Board of Fire Underwriters* require an approved manually-reset thermostatic device set at a maximum of 125 F, which is arranged to shut down the fans automatically when the temperature of the air in the system becomes excessive, as from fire. While there is such a device used principally in connection with domestic, forced draft, hot air heating systems, there is no device meeting these requirements which is now listed by Underwriters' Laboratories, Inc. The present regulations covering the protection of walls and partitions take care of the specifications covering

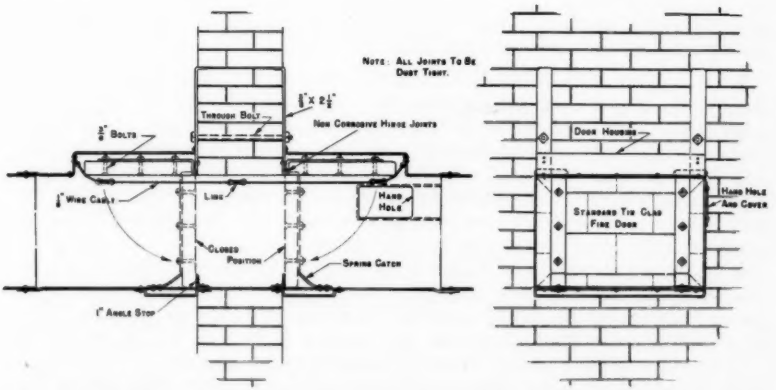

FIG. 3. SUGGESTED TYPE OF AUTOMATIC HINGED FIRE DOORS FOR DUCT PASSING THROUGH OPENING IN FIRE WALL

fire cut-offs in ducts, but there is no reason why there will not be special designs of fire doors and dampers which will be submitted to the Laboratories for testing and listing. The fire resistive qualities of duct linings may also be investigated, as well as extinguishing agents of different types, as to their practical application with respect to air conditioning. Some form of dependable smoke detection will undoubtedly be investigated. At the present time, the only reference to such protection is contained in a note printed in the August, 1938, amendment to the rules, which reads as follows:

It is recognized that in certain occupancies there is a potential panic and smoke damage hazard. For such conditions it is recommended that consideration be given

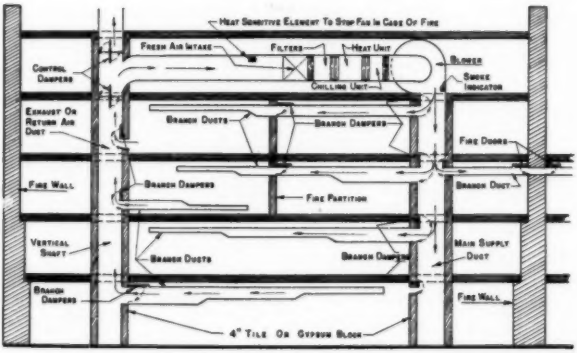

FIG. 4. TYPICAL INSTALLATION OF AN AIR CONDITIONING SYSTEM IN A BUILDING OF FIREPROOF, SEMI-FIREPROOF, OR HEAVY MILL CONSTRUCTION

to the development and application of suitable smoke detectors to automatically shut down the system and cause an alarm signal.

Several years ago when air conditioning began to really develop, it was thought that the filter was the disturbing factor as far as the fire hazard was concerned. Then fires in combustible duct linings called attention to the fact that ducts should be of incombustible construction. But the latest development and the fire which occurred in a Detroit department store showed that still other factors such as combustible debris in the duct system might add to the panic and smoke loss. A new system can be so designed that this smoke hazard is reduced to a minimum but, in connection with those not so designed and old systems not in compliance with the rules, some form of smoke detection will undoubtedly be required which will operate at sufficient speed to prevent smoke and hot gas from being discharged into buildings housing occupancies where the panic hazard is pronounced. Although the protection of each air conditioning system is a special engineering problem as the designs generally differ in some respect, the purpose of the smoke detection system would be to shut down the fan and close dampers, preventing the smoke from entering the building and discharging it through a suitable opening, the fire

being controlled by some extinguishing agent not connected with the smoke detecting apparatus.

Although the developments in the air conditioning industry are advancing rapidly, it can readily be seen that many of the necessary regulations are at present in a state of flux. The air conditioning profession and industry are cooperating in the development of suitable protection, for they realize that if acceptable safety features can be incorporated in these systems it will eliminate the possibility of seriously retarding the progress of the entire industry. Proper design and maintenance will materially assist in obtaining an acceptable installation which can be suitably protected.

Due to the inherent features of an air conditioning system, it is impracticable to suggest specific fire protection that would properly apply to all cases, and therefore it is necessary to give each system requiring protection special study and treatment. Most systems, and especially the smaller ones and those confined to one floor or part of a floor, will not usually require any special fire protection if installed according to the regulations, but the large systems supplying more than one floor from below may present a hazard which should be protected, regardless of installation methods. It then follows that the best assurance against loss by fire is not only a standard installation, but cleanliness at all times, good maintenance, and constant vigilance by the owner or occupant.

DISCUSSION

H. C. MURPHY (WRITTEN): This paper brings out a number of questions which are of much interest and concern to heating, ventilating and air conditioning engineers.

The opinion has been expressed by some of the outstanding engineering authorities in the *National Fire Protection Association* that there has been considerable hysteria in considering the possible fire hazards involved in air conditioning. There is undoubtedly some basis for this opinion. However, in an industry which has grown as rapidly as air conditioning, there probably will be a certain number of manufacturers and installers who either from ignorance or from the urge to secure business, install improper equipment or install proper equipment improperly.

Whether we like it or not we will have regulations governing the manufacture and installation of air conditioning equipment. If some outstanding fire or catastrophe, similar for instance to the Iroquois theater fire, should be chargeable to a poorly designed or poorly installed air conditioning system the whole industry would suffer, the innocent with the guilty. The public would not differentiate between poor equipment and good equipment.

To my mind, the Society would be well advised to cooperate actively with the *NFPA* in the development of a nation wide code, rather than be subjected to hundreds of different city or state codes developed by local fire marshals and fire departments, the requirements of which may vary from city to city and state to state. I have personally found the Blower Systems Committee of the *NFPA* entirely reasonable and disinclined to set up unduly restrictive requirements. Practically all of the members of the Committee are engineers and 6 of the 18 are members of the ASHVE. I have found the Committee amenable and open-minded to any argument which has a practical engineering background.

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The author covers various phases of air conditioning. I would like to discuss particularly some of the problems of air cleaners which he has brought out.

It is, I find, a very general opinion of engineers who have studied this subject, that properly designed air filters can be a definite fire protective measure and it certainly appears that only under conditions of the most gross carelessness and flagrant disregard of the present regulations of the *NFPA* that a filter could be ignited.

It is standard practice with manufacturers of large electrical equipment, turbo generators, mill motors and the like to insist upon the protection of their equipment by air cleaners. They have found by experience that even a thin coating of soot or similar carbons over the windings acts as a heat insulating medium and prevents the proper dispersion of heat, sometimes causing dangerous overheating and fires. A typical case is that reported in the April 1938 quarterly of the *National Fire Protection Association,* in which a motor in an air conditioning installation in a bank building at Shreveport, La., caught fire from overheating.

There are, in fact, many conditions where it is standard practice to eliminate possible fire hazards with air filters, as they unquestionably prevent the spread of fire, confining it to spaces near the air intake where it can be more readily combatted instead of allowing it to be fed through the ducts from room to room and floor to floor by the dust accumulated in the system. The records of the National Board of Fire Underwriters do not show a single instance where the fire progressed beyond the air filter in a system thus protected. The fire burned the dust in the duct in front of the air filter and in some cases burned the filter itself, but the fire itself stopped there as there was no dust deposited in the duct in back of the filter to lead the fire further.

It is, of course, entirely possible to manufacture fireproof air filters and there is very little excuse, except on a price basis, for the continued use of untreated filters made of cardboard, paper, hair, sisal, Spanish moss and the like, although many thousands of such filters are in every day use. However, regardless of the inflammability or non-inflammability of the filter itself the dust, as pointed out by the author, will burn whether it is on the face of a non-combustible air filter or distributed throughout the duct system. It is obviously safer to collect this dust at the entrance to the system on the face of a non-combustible air filter, where it can be readily reached, than to allow these inflammable coatings throughout the entire duct system.

The author gives some details of the test procedure developed by the Underwriters' Laboratories for testing air filters. It is of interest to know that the Underwriters' Laboratories found it impossible to test filters at the standard air velocity recommended by all air filter manufacturers as the air current quickly extinguished the fire. It was only by adopting a much lower air velocity that most filters could be burned in the test set-up and a comparison of inflammabilities could be secured. Until November 1938 no air filter had received the approval of the Underwriters' Laboratories. There are now 8 different types manufactured by one company, and 2 types produced by another manufacturer which carry the label of the Underwriters' Laboratories as to compliance with the regulations of the National Board of Fire Underwriters. These filters consist of 3 different types of media: thin sheets of flexible glass filaments or fireproofed cellulose sheets, spun glass pads and all metal filters coated with so-called filter oils.

In the Detroit department store mentioned by the author, the records of the National Board of Fire Underwriters show that an improper filter oil was used. If commercially available charging liquids conforming to the regulations of the National Board of Fire Underwriters had been in use, the fire could not have occurred.

Many carefully designed and properly installed air cleaners are turned into potential fire hazard by the substitution of improper oils for the charging liquids recommended by the air filter manufacturer. Many of the materials freely offered by the oil companies for this purpose are distinctly inflammable and in no way meet the requirements of the National Board of Fire Underwriters for filter charging liquids. The author states "The ultimate aim of the air filter manufacturer is to discover a liquid which will not burn and at the same time possess all the filtering qualities of oil." Such materials have been developed, and approved by the Underwriters Laboratory for use with certain viscous filters.

In the second department store fire in which the author found lint, dust, gum wrappers and similar material distributed in the duct in back of the air filter, it is only fair, I believe, to point out that the air filter in question had not been in operation for many months prior to the fire.

I believe the author has presented a valuable contribution to our knowledge of possible fire hazard in connection with air conditioning systems. I believe further that by careful study and cooperation we can produce, without prohibitive expense, systems which will not introduce any additional fire hazard.

Until recently the Society has not, to my mind, given adequate attention to the work the Fire Underwriters are doing. I believe that if there are any unduly restrictive requirements in the present code, they can be eliminated if the matter is brought to the attention of the Committee in adequate engineering form.

When city fire ordinances or state fire ordinances are set up they are frequently developed by men who are not engineers and who may not have a proper understanding of the engineering problems entering into the production of equipment. There are, in addition, sometimes political factors which make it almost impossible to arrive at practical solutions to the problems.

ERNEST SZEKELY (WRITTEN): It is impossible to comment on this paper in any but general terms and then it becomes necessary to repeat what has been so often stated. The author makes a good presentation of his subject and should be commended for his efforts, but we, who are actively interested in air conditioning, are forced to feel that the fire protection engineers have become unduly panicky about hazards produced or likely to be produced by air conditioning installations. This probably explains why rules and regulations are being promulgated which, if accepted, will make the first cost of air conditioning systems prohibitive and thus may be the source of hindering healthy development of a new industry. We feel particularly keenly on this point because our observations do not indicate that air conditioning systems themselves have contributed greatly to the fires reported.

The modern air conditioning system from viewpoint of fire hazard is very little different from heating and ventilating systems of former generations. New elements have been introduced with the wide spread use of filters and refrigeration coils. Anything that can be done for safety with these two new factors should be done and we assume that the Underwriter's Laboratories are engaged in testing and developing proper materials for protection and preparing specifications to prevent the use of hazardous equipment and materials.

Judging from various discussions of systems in this paper, it would appear that the fire protection engineers consider the duct system as representing a great potential danger. Ventilation systems used in schools, involving multiple branches both in supply and exhaust systems, distributing air over a number of rooms and several floors do not seem to have a record of being in any way responsible for fires or promoting the propagation to any material extent.

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The 11 points the author proposes are worthy of consideration and probably no one will deny their necessity or reasonableness but we are very much concerned about the manner of execution of these provisions as it will be very harmful to the air conditioning industry to introduce a great many protective measures which could not be justified by the principles of safety and at the same time would complicate the air conditioning business.

I for one am fully convinced that points 10 and 11 really represent the most vital factors of safety and should be pushed vigorously by all concerned.

Some method of detecting smoke, I believe, may be worth studying so that possibility of panic may be prevented, in densely occupied spaces. It is important, however, that such a device should distinguish between smoke and other sources of haziness, which may be quite harmless in nature.

In conclusion, I should like to point to the report of the Wisconsin Chapter.[1] This report specifically expressed objection to rules proposed in pamphlet 90 and in my mind these objections should be seriously considered by anyone interested in this matter.

A. J. HESS (WRITTEN): The fact that fire protection in air conditioning systems is receiving consideration at a Semi-Annual Meeting of the Society is the most interesting and important part of the paper under discussion.

Consulting and design engineers have been in the dark long enough on this important subject because, for some unexplainable reason, the Underwriters and other inspection groups have failed to pass on to these engineers the results of field experiences and laudable research on this subject. That Mr. Loughead recognizes this situation is a step in the right direction. Let us hope there are many more steps in this same direction for, if the designer can be so educated that his designs will have a minimum of fire hazards, the chances of dangerous and expensive fires from these systems will be minimized.

Air filters have always been branded as the number one fire menace in air conditioning systems; this has always seemed rather far fetched to the air conditioning engineer for several reasons. In the first place, the removal of dust from the air entering a duct system is a very important move in the prevention of the spread of fires through duct systems. It is obvious that a fire cannot spread through ducts without fuel, and dust is usually the only fuel available so it can be seen that by removing dust from air, filters act as fire control devices instead of fire menaces. Further, most fire prevention ratings made of filters by inspection groups are based on clean filters when obviously no filter under operating conditions can be clean if the filter is doing its intended work. Of course, proper maintenance of filters will keep this concentration of dust to a minimum and thus reduce any hazard that exists at the filters due to dust. In passing it should be pointed out that the maligned automatic oil filter is automatically and continuously cleaned and has a minimum of dangerous accumulated dust, and further, this process is not dependent on the vagaries of some maintenance man, as is the case with all other filters. It follows therefore that filters should be regarded as an ally of fire protection equipment and that the fire hazard of filters exists principally in the accumulated dust which is, within reasonable limits, an inherent necessity of filtration and the control of filters as fire hazards lies in an educational campaign for proper maintenance. The best recommendation an engineer can make to provide maximum fire protection is the installation of some form of automatic sprinkler regardless of whether the filter itself is fireproof or not.

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[1] Discussion on "Effect of Size and Type of Air Inlet and Outlet on the Heat Output of Convectors," Vol. 45, 1939, p. 544.

The subject of duct materials and duct lining merits much more than a mere mention, as the experience of several major fires has proven. Modern design of air conditioning systems calls for the absorption of any sounds which might be transmitted through ducts to the conditioned spaces and insulation of duct walls against thermal losses. Most materials which are good for sound absorption are also good thermal insulators and hence there is a natural tendency on the part of designers, with an eye to economy, to either construct the ducts of or to line metal ducts with materials which will simultaneously insulate thermally and isolate acoustically. This is good engineering practice, but an intense educational campaign should be instituted to teach engineers to use only fireproof materials for this work. Past experiences show that fires will spread very rapidly throughout a building if duct lining or the ducts themselves can burn, for such a duct usually acts as a flue even though the fans may be inoperative.

From a discussion of duct material and duct lining it is only a short and logical step that a fire prevention study be made of the use of parts of non-fireproof buildings as ducts, such as spaces above corridor ceilings and attics. In many installations where first cost seems to be the only consideration, the use of such spaces as ducts is often made. There are few reasons which would cause a fire to spread through a structure more rapidly than such plenum spaces constructed of combustible material and under the forced draft of a fan. The rapidness of destruction, with the attendant loss of life, of the dormitory at Boulder Dam which burned a few years ago was probably caused by such construction even though the fire itself was proven to originate otherwise. Definite rules and regulations should be established for such cases as this, and proper ordinances should be encouraged.

The author touched on the use of spaces in buildings as plenums, but the subject is of more importance than the space allotted it. Modern research has established the fact that in most jobs a great percentage of the air conditioning cooling load is due to radiant heat from the sun (this is especially true in one story buildings). The heat from the sun has many short wave components that pass through materials of roofs into attic spaces where, due to an increase in the wave length, the heat is trapped and raises the temperature of the attic space far above the outside temperature. Engineers have found that by pumping air through such spaces this excess heat can be removed economically and easily. There is nothing wrong with the practice, but to date very few such jobs have any safeguards against fire in these spaces which are dangerous because they are seldom, if ever, inspected and under the described conditions can spread fires through the roof spaces and into the ducts which terminate there. Rules governing fan shutoff and fire dampers should be worked out for this problem. Each year the volume of attic fans sold is getting greater, especially in residences, and each attic fan means one more of this type of potentially dangerous installation unless properly safeguarded. Installations which exhaust air from the air conditioning system through attic spaces also come under this head as do systems using the basement as a plenum source of cool air. It has been of interest to the writer that the Underwriters have not taken steps in this matter before now.

Areas in which gas is the major fuel also have a pressing problem unmentioned in the paper. This problem has to do with the location of forced air furnaces especially with reference to fans and in cases where refrigeration is used with reference to dehumidifying cooling units. The furnaces normally used in gas burning districts are many times of rather sleazy construction and usually do not have an element properly resistant to corrosion. If the furnace is located on the suction side of the fan, it is quite often possible to draw the fire and products of combustion through a defective heating element, which has corroded, into the conditioned air stream being pumped into the building. The ease of automatically controlling gas appliances aggravates the situation because inspections are very infrequent since they seem unnecessary to the layman. Such installations are fire hazards and incidentally carbon-monoxide

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hazards. Where refrigeration cooling units are on the suction side of a furnace, the rate of corrosion of the furnace is increased due to corrosion inside the furnace element caused by condensation of moisture from the outside air drawn down the flue from the outside. This merely aggravates the situation mentioned if the furnace is on the suction side of the fan.

This whole situation of fire protection and control is very important and merits study and research on the part of the Society or at least an effort on our part to cooperate with the Underwriters in their work on this subject. We need their knowledge of fire prevention and they need our knowledge of air conditioning systems; between the two groups it is possible that a good practical, safe, and sane set of rules and regulations can be promulgated which will afford owners of air conditioning systems practical fire protection at reasonable cost, and which will not adversely affect good design, installation or maintenance. Further, such cooperation may convince the Underwriters that one of the easiest ways to get general public acceptance of fire protection recommendations in air conditioning systems is to have these recommendations incorporated into designs of outstanding installations.

PROF. C. M. HUMPHREYS: This paper discusses a subject which is of interest to all of us, and the author should be commended for presenting it to us. There is one detail however, which I believe should be changed.

In Fig. 3 of the paper, the author shows an arrangement of automatic hinged fire doors to be installed in a duct passing through a fire wall. Both doors are held open by a single ⅛ in. wire cable in which three fusible links, one in the center and one near each end, are inserted. With this arrangement, if the center link in the cable melts first the doors will close. If either of the end links should melt before the center link, the wire cable will hang across the 1 in. angle iron frame and thus prevent tight closure of the door.

If a double fire door is needed in a duct where it pierces a fire wall, then the design of the support should be such that positive closure of both doors is assured. It should also be noted that the relation of duct depth to wall thickness will effect the safety of the device shown in Fig. 3. Fire door supports recommended by the Fire Underwriters should be positive in action for any combination of duct and wall dimensions.

J. B. HEWETT: I had experience with two fires each amounting to approximately $1500.00 damage to an air conditioning system in a newspaper plant. An investigation of the cause for the fires was made at the request of the company holding the fire insurance for this installation. It was discovered that the dirt collected on the filters was found to be inflammable and the movement of the air through the filters resulted in a temporature sufficiently high to melt silver solder on the direct expansion coils located within 3 ft of the filter bank.

The damage caused by the fires was confined to the air conditioning equipment, but smoke was circulated throughout the building since the fan continued to operate even though the current of the fan was shut off. A fire damper with a fusable link set to operate at a comparatively low temperature was located immediately after the filter bank to stop the high velocity air movement through the filters and reduce the circulation of smoke through the system.

A representative of the New York Board of Fire Underwriters insisted the filters be changed to a type found to be less inflammable. However, as a result of the investigation which we had conducted it was my contention that the type of filter used was not important since the dust was inflammable.

No. 1136

SUMMER COOLING REQUIREMENTS IN WASHINGTON, D. C., AND OTHER METROPOLITAN DISTRICTS

By F. C. HOUGHTEN,* (*MEMBER*), CARL GUTBERLET ** AND ALBERT A. ROSENBERG,***
PITTSBURGH, PA.

SINCE 1935 the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS has been studying the cooling requirements of persons seated at rest or engaged in sedentary occupations in summer cooled and air conditioned spaces. During the summers of 1935 and 1936, laboratory studies of trained subjects were made by research men in Pittsburgh, Toronto, and College Station, Texas. The findings in these studies [1, 2] showed that for persons in any geographical region, the effective temperature scale applied without any very pronounced independent effects of relative humidity over a fairly wide range. Little variation in the indoor summer cooling requirements was found for Pittsburgh and Texas, having maximum mean normal summer temperatures of 74.5 and 84.0 F, respectively. However, for Toronto, having a considerably lower maximum mean normal temperature of 68.2 F, the subjects showed a desire for approximately 2½ deg lower effective temperature.

During the summer of 1937 laboratory studies were continued at College Station, Texas, and Toronto, Canada, while a fairly comprehensive field study was made of the requirements of 275 office workers in a summer cooled and air conditioned space in Minneapolis, Minnesota. The Texas and Toronto studies [2] more or less verified the findings of the previous year.

The Minneapolis study,[3] besides being very conclusive in regard to the requirements of the office workers in that city, indicated some variation in the requirements of different geographical regions. As a result, three studies

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* Director, ASHVE Research Laboratory.
** Research Assistant, ASHVE Research Laboratory.
*** Research Engineer, ASHVE Research Laboratory.

[1] ASHVE RESEARCH REPORT No. 1035—Comfort Standards for Summer Air Conditioning, by F. C. Houghten and Carl Gutberlet. (ASHVE TRANSACTIONS, Vol. 42, 1936, p. 215.)

[2] ASHVE RESEARCH REPORT No. 1055—Cooling Requirements for Summer Comfort Air Conditioning, by F. C. Houghten, F. E. Giesecke, C. Tasker and Carl Gutberlet. (ASHVE TRANSACTIONS, Vol. 43, 1937, p. 145.)

[3] ASHVE RESEARCH REPORT No. 1088—Summer Cooling Requirements of 275 Workers in an Air Conditioned Office, by A. B. Newton, F. C. Houghten, Carl Gutberlet and R. W. Qualley. (ASHVE TRANSACTIONS, Vol. 44, 1938, p. 337.)

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Mackinac Island, Mich., July, 1939.

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similar to that made in Minneapolis during 1937 were carried out during the summer of 1938: The results of one of these studies, in San Antonio, Texas have already been published;[4] the data from a second study, conducted in the air conditioned offices of the Metropolitan Life Insurance Building, New York, have not yet been completely analyzed and will be reported at a later date. This present report deals specifically with the results of the study conducted in the air conditioned offices of the Federal Reserve Building, Washington, D. C., with some comparison of the results of all the previous studies.

Through the cooperation of the building management a number of offices in the modern, air conditioned Federal Reserve Building in Washington were made available for the study from August 1 to August 31, 1938. These offices were typical of the rest of the offices in the building, and included 73 workers of both sexes, of wide age distribution and rank in the organization.

Throughout a study during any certain day a predetermined effective temperature was maintained through the operation of the central air conditioning

TABLE 1. DAILY REPORT RECORD

(Thursday, August 4, 1938, Sheet No. 1)

CASE NO	TIME	INSIDE AIR CONDITIONS				PM LEAVING REACTION	AM ENTERING REACTION	CURRENT FEELING S.C. – M.C. C MW – S.W.	PERSPIRATION FOREHEAD	CLOTHING	AGE	SEX	OUTDOOR CONDITIONS TIME OF ENTRY 8 AM		TIME OF LEAVING 3 PM	
		WB	DB	ET	RH								DB	WB	DB	WB
13	9:52	63	75	70.1	52	NO	NO	C	N	L	25	F	84	77.5	87	79
	2:00	64	75	70.5	56			C								
14	9:52	63	75	70.1	52	NO	NO	C	N	L	45	F	84	77.5	87	79
	2:00	64	75	70.5	56			C								
15	9:54	63	75	70.1	52	NO	NO	C	N	L	29	M	84	77.5	87	79
16	9:56	64	77	71.5	50	YES	NO	C	N	L	44	F	84	77.5	87	79
17	9:58	64	79	72.5	44	NO	NO	MW	N	L	30	M	84	77.5	87	79
	2:03	63	77	71.2	46			C								
18	10:00	64	79	72.5	44	YES	NO	C	N	L	41	F	84	77.5	87	79
	2:05	64	78	72.0	47			C								

system, while a trained observer recorded the atmospheric and personal data. The data collected included a record of outside-dry-bulb and wet-bulb temperatures, wind velocity and other weather conditions. To obtain inside conditions, the observer carried a motor-driven psychrometer which he placed in a location construed to be representative of air conditions in that room or portion of the room. This instrument was allowed to adjust itself to equilibrium while the occupants were interrogated, and after the elapse of sufficient time the dry- and wet-bulb temperatures were recorded; the data of the room conditions and answers of the occupants were recorded on a form similar to Table 1.

Previous to the actual study, the 73 workers participating were given case numbers, and their individual characteristics, including age or approximate age, stature, weight, apparent health, etc., were recorded for future reference. As shown in Table 1, the following observations were made: Time of day at which the observation was made; the dry-bulb and wet-bulb temperatures

[4] ASHVE RESEARCH REPORT No. 1127—Reactions of Office Workers to Air Conditioning in South Texas, by A. J. Rummel, F. E. Giesecke, W. H. Badgett and A. T. Moses. (ASHVE TRANSACTIONS, Vol. 45, 1939, p. 459.)

representative of air conditions in which the worker was present; the worker's reaction upon leaving the cooled space that afternoon (recorded the day following); his reaction to entering the cooled space that morning; his current feeling of warmth and degree of perspiration; and the classification of his clothing as to light, medium, or heavy weight. From the dry-bulb and wet-bulb temperature readings the relative humidity and the effective temperature were later determined and recorded.

The same index of feeling of warmth employed in earlier studies was used. According to this index, a feeling of *ideal comfort* is recorded as *C* or given

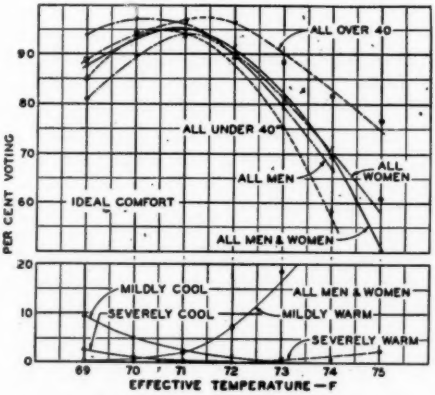

FIG. 1. RELATION BETWEEN EFFECTIVE TEMPERATURE AND THE PER CENT OF OFFICE WORKERS IN WASHINGTON INDICATING IDEAL COMFORT. CURVES FOR ALL MEN AND WOMEN, ALL MEN, ALL WOMEN, ALL OVER 40, AND ALL UNDER 40. PERCENTAGE CURVES IN LOWER CHART INDICATING OTHER FEELINGS OF WARMTH ARE FOR ALL MEN AND WOMEN

the number "4"; comfortably cool and comfortably warm are indicated as *MC* and *MW*, and numbered "3" and "5," respectively; feelings of severely cool and severely warm are indicated as *SC* and *SW*, and numbered "2" and "6," respectively; while decidedly cold and decidedly hot are numbered "1" and "7," respectively. In this study, however, only "2" to "6," inclusive, were used.

Degree of perspiration was indicated in accordance with the index used in earlier laboratory studies; that is, "0" for forehead or body, dry; "1" for forehead or body, clammy; "2" for forehead or body, damp (perspiration just visible); "3" for forehead or body, wet (sweat covering the surface, frequently in drops); "4" for perspiration on the forehead running down, or perspiration on the body running down and wetting the clothing. In this study, however, most of the observations were taken after at least one hour's occu-

pancy in the cooled space, and for this condition there was usually no perspiration, which is indicated by the letter *N*.

The data were collected by employees of the Federal Reserve Board and were sent to the Research Laboratory of the Society for analysis. In analyzing the data, all observations made on all workers present on all days during the study were tabulated according to the effective temperature pertaining with relation to the comfort index. The total number and, from that, the percentage of the total number of observations for each comfort index at each degree effective temperature was tabulated; from these tabulated data the curves in Figs. 1 and 2 were plotted. In these charts *ideal comfort* curves, or index

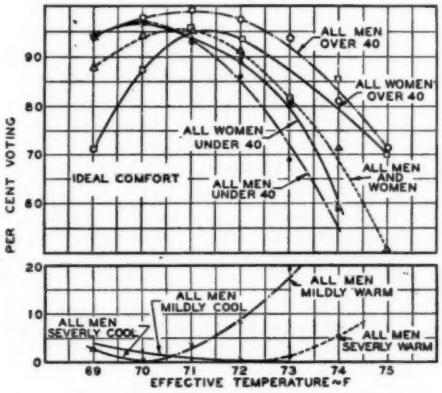

FIG. 2. RELATION BETWEEN EFFECTIVE TEMPERATURE AND THE PER CENT OF OFFICE WORKERS IN WASHINGTON INDICATING IDEAL COMFORT. CURVES FOR ALL MEN AND WOMEN, ALL MEN OVER 40, ALL MEN UNDER 40, ALL WOMEN OVER 40 AND ALL WOMEN UNDER 40. PERCENTAGE CURVES IN LOWER CHART INDICATING OTHER FEELINGS OF WARMTH ARE FOR ALL MEN AND WOMEN

"4," are plotted for all workers taking part in the study, all workers over 40 years of age, all workers under 40 years of age, all men, and all women. It will be observed that in each case a maximum-distribution curve is indicated, showing a maximum percentage of comfort at some definite effective temperature, with a falling off in percentage in either direction. In order not to subject the workers to undue extremes of low and high temperatures, most of the observations were limited to the effective temperature range of 69 to 75 F; while a few observations were made below and above these effective temperatures, the number were too few to give accurate indications. Hence, only a small portion of ideally comfortable curves are drawn, differing in this respect from some of the curves in the earlier reports. However, the main objective

of this study was to determine the temperature range over which the maximum percentage experienced comfort, rather than a study of conditions where the percentage of comfort was small and the percentage of discomfort was great.

In general, the men indicated a maximum percentage feeling comfortable at a slightly lower temperature than women; while workers over 40 years of age, either for the combined group or for either sex, show an optimum percentage of comfort at a slightly higher temperature. One characteristic difference in the results of this study from the one made in Minneapolis during 1937 is the high percentage indicating *ideal comfort* for any group. This

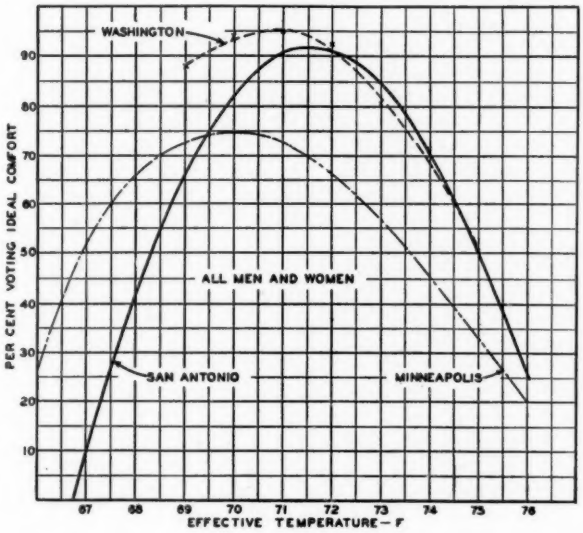

FIG. 3. RELATION BETWEEN EFFECTIVE TEMPERATURE AND THE PERCENTAGE OF PERSONS INDICATING IDEAL COMFORT. CURVES FOR ALL MEN AND WOMEN IN WASHINGTON, SAN ANTONIO AND MINNEAPOLIS

is undoubtedly a result of the difference in method of collecting the data. In the Minneapolis study, daily questionnaire cards were given to the individual workers to fill out, while in the Washington study a trained observer obtained and recorded the information. This resulted in more consistent replies for this study; and also, the direct observation of the wet- and dry-bulb temperatures in the immediate vicinity of the worker probably gave a better indication of the true atmospheric condition.

The expressions of feeling of comfort "2" and "3" and "5" and "6" are also plotted in Figs. 1 and 2, for all men and women and for all men, respectively. The same general characteristics for these curves are indicated as

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were found in the earlier studies; that is, a minimum of discomfort results at approximately the same effective temperature as maximum comfort, with a slightly higher percentage indicating mildly cool than those indicating mildly warm for the condition of optimum comfort.

Data on the shock experienced by the workers upon entering the cooled space in the morning and the reaction upon returning to the hot outside during the afternoon were collected. While there were a few indications of shock or reaction, the number never exceeded 2 per cent, and this small percentage showed no consistent relationship to either the effective temperature main-

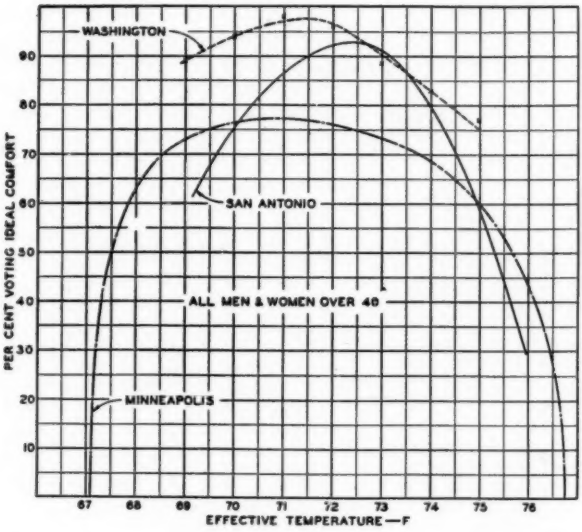

FIG. 4. RELATION BETWEEN EFFECTIVE TEMPERATURE AND THE PERCENTAGE OF PERSONS INDICATING IDEAL COMFORT. CURVES FOR ALL MEN AND WOMEN OVER 40 IN WASHINGTON, SAN ANTONIO AND MINNEAPOLIS

tained indoors or the outside temperature before entering or after leaving. This study therefore adds nothing to this phase of the subject beyond the conclusions resulting from the previous Minneapolis study.[5]

It is of interest to compare the results of this study with the other studies by or in cooperation with the Society's Laboratory since 1935, and this is done for the different worker or subject groups in Figs. 3 to 7, inclusive. Curves of *ideal comfort* are drawn for each metropolitan district from which data were available, but in comparing these curves it should be emphasized that

[5] ASHVE RESEARCH REPORT No. 1102, Shock Experiences of 275 Workers After Entering and Leaving Cooled and Air Conditioned Offices, by A. B. Newton, F. C. Houghten, Carl Gutberlet, R. W. Qualley, and M. C. W. Tomlinson. (ASHVE TRANSACTIONS, Vol. 44, 1938, p. 571.)

the maximum percentage indicating ideal comfort in any locality is probably affected by the method of collecting the data more than from the difference in actual feeling of comfort. The effective temperature at which optimum comfort occurs, however, should be largely due to the difference in geographical regions, possibly with some variations due to lack of or imperfection in imparting to the persons interrogating, the truth of what should be construed as ideally comfortable. However, it is probable that this factor is not of great importance. The curves in these charts show a lower temperature as desired by the Minneapolis and Toronto workers or subjects, with relatively little

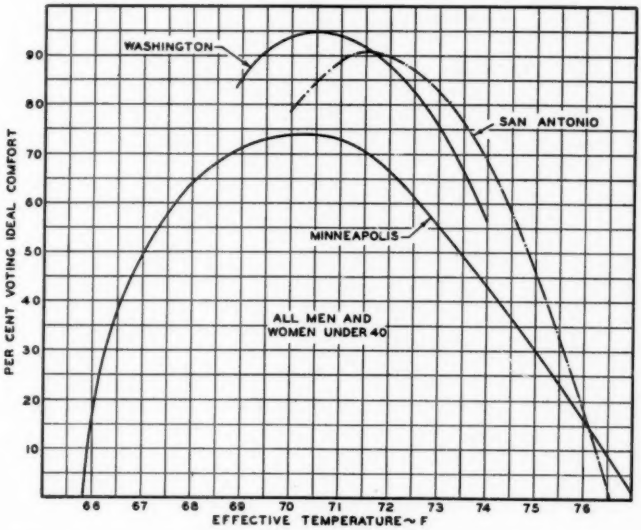

FIG. 5. RELATION BETWEEN EFFECTIVE TEMPERATURE AND THE PERCENTAGE OF PERSONS INDICATING IDEAL COMFORT. CURVES FOR ALL MEN AND WOMEN UNDER 40 IN WASHINGTON, SAN ANTONIO AND MINNEAPOLIS

variation due to geographical location in other instances, where the same type of individual is concerned.

A review of the data plotted in Figs. 1 to 7, inclusive, gives a fairly concrete picture of cooling requirements for summer air conditioning for widely scattered metropolitan districts. The data collected in Washington last summer and plotted in Figs. 1 and 2 indicate that about 95 per cent of all workers in the Federal Reserve Building during the period from August 1 to August 31 were comfortable at 71 ET. If a tolerance of plus or minus one degree is allowable a minimum of 91 per cent was comfortable. These facts are of much greater practical significance than the small variations in effective temperature for the maximum percentage of comfort shown for different age and sex groups. As

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an example, compared with about 95 per cent comfortable at 71 ET for all groups, about 97 per cent of those over 40 years of age were comfortable at 71½ ET, with about the same percentage of comfort for all men at a little above 70 ET, and for all men and women under 40 the percentage was 95 at 70½ ET. While these differences are of some significance when considered from an academic point of view they cannot be given much consideration in practical application of air conditioning. Further, it is of interest to note that the optimum percentage for all curves falls well within the range of 70 to 72 ET, again indicating that a practice of maintaining 71 ET with a

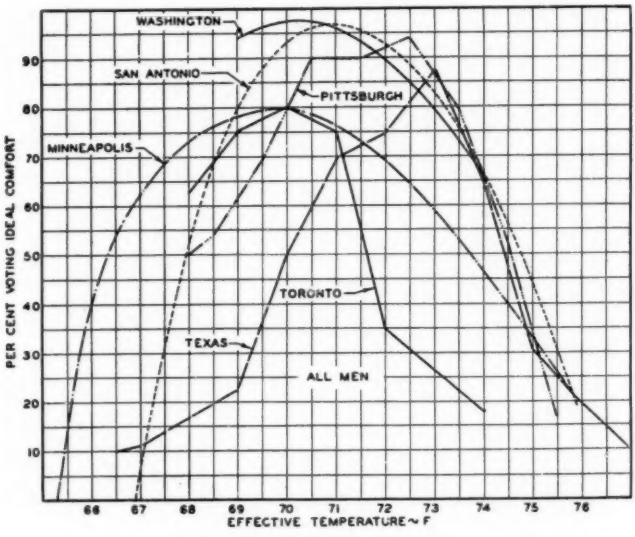

FIG. 6. RELATION BETWEEN EFFECTIVE TEMPERATURE AND THE PERCENTAGE OF PERSONS INDICATING IDEAL COMFORT. CURVES FOR ALL MEN IN WASHINGTON, SAN ANTONIO, MINNEAPOLIS, PITTSBURGH, TORONTO AND TEXAS

tolerance of plus or minus one degree would satisfy all groups. If an operating tolerance of 2 deg were required it would reduce the percentage of comfort of any group to approximately 70 per cent. It is also of interest to note that the minimum percentage of those feeling either mildly or slightly warm or cold also comes within this temperature range.

When viewing the results of the several studies in different metropolitan districts, one is confronted with a greater variation. Inspection of the curves shows that they more or less fall into two groups, the Minneapolis and Toronto data showing consistently a desire for lower temperatures than those for the other cities. All curves for Minneapolis and Toronto show maximum percentages at about 70 ET with variations in Minneapolis for all women,

and all men and women over 40 of 70½ and 71 ET, respectively. Curves for the metropolitan districts other than Toronto and Minneapolis show maximum percentages at effective temperatures ranging from about 70½ for all men and women under 40 in Washington, to 73 deg for all men in the College Station, Texas, study. However, it should be pointed out that the broken-line curves in Fig. 6 for Pittsburgh and College Station, Texas (this was also the case for the Toronto study), resulted from studies on relatively few laboratory subjects, giving insufficient data for a statistical analysis resulting in smooth curves. Judging from the nature of the curves it is probable that statistical data for Pittsburgh and College Station, Texas, would have given

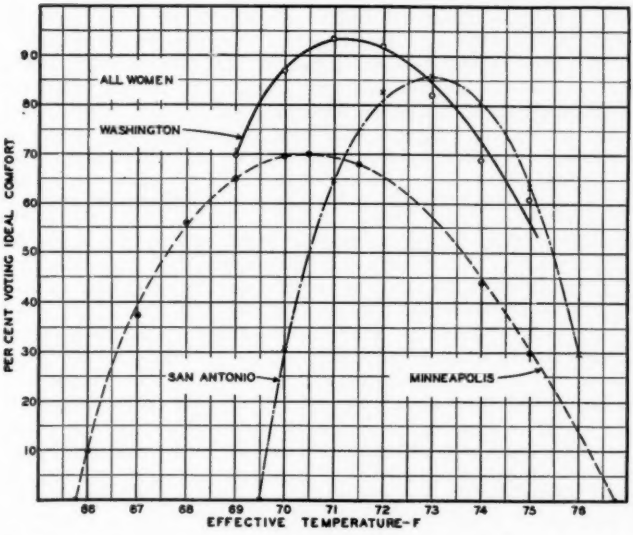

FIG. 7. RELATION BETWEEN EFFECTIVE TEMPERATURE AND THE PERCENTAGE OF PERSONS INDICATING IDEAL COMFORT. CURVES FOR ALL WOMEN IN WASHINGTON, SAN ANTONIO AND MINNEAPOLIS

maximum percentages voting comfortable at about 71½ and 72½ ET, respectively, which would reduce the effective temperature spread to from 70½ to 72½ deg, or a total spread of 2 deg.

In this connection, if one were to accept 71½ deg as the best single practical effective temperature to have been maintained in all of these instances, it would have given percentages of comfort of about 72 for all men at College Station, Texas, 90 for all men at Pittsburgh, 95 for all men at San Antonio, and 93 for all men at Washington; 75 for all women at San Antonio and 93 for all women at Washington; 90 for all men and women over 40 at San Antonio, and 97 for all men and women over 40 at Washington; 91 for all men and women under 40 at San Antonio, and 92 for all men and women under 40 at Washington; 92 for all men and women at San Antonio, and 94 for all men

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and women at Washington;—compared with maximum percentages for the same groups and locations of: 87, 94, 97, 97, 86, 93, 93, 97, 91, 95, 92, 95, respectively. With the exception of the men studied at College Station, Texas, and the women studied at San Antonio, the percentages which would be adversely affected by maintaining a common condition of 71½ ET in all locations would be insignificant.

It is of interest to compare these deductions with the normal mean summer temperatures for some of these localities. Fig. 8, taken from an earlier Laboratory publication,[6] gives such data for Boston, Toronto, Pittsburgh, and College Station, Texas. It will be noted that the mean summer temperature in Toronto is considerably lower than those for Pittsburgh and San Antonio, Texas. It is probable that the normal mean temperature in Minneapolis would fall more nearly around that of Toronto, while that of Washington would probably fall between that of Pittsburgh and San Antonio. This might indicate that localities having a maximum normal mean summer temperature of 74 F or above should maintain an effective temperature of about 71 deg, while those localities showing normal mean summer temperatures of below 74 F should maintain effective temperatures below 71 deg. Beyond this, some adjustment, probably not to exceed one degree effective temperature, might be based upon peculiar local conditions or peculiarities of the type of occupants.

It should be emphasized that the data contained in this report and the deductions offered previously apply to persons entering the cooled space and remaining for three hours or more, which may be construed as permanent occupancy; that is, the occupants remain in the cooled space long enough to become in equilibrium with it. There are reasons for questioning the application of these results to occupancy of a shorter period. The shock data published in earlier reports[7] indicated that maintaining the suggested effective temperature results in no serious shock, and it should be emphasized that this same shock will be experienced whether a person enters the cooled space for 5 min or for permanent occupancy. Hence, the same conditions as are maintained for permanent occupancy should give satisfactory results for occupancy of shorter duration. However, for shorter periods of occupancy the people may be reasonably satisfied with the sensation of coolness acquired during the short period of time. Therefore, it may be reasoned with some justification that while for a short period of occupancy maximum comfort may be expected in the same conditions as are maintained for permanent occupancy, the occupants may be as well satisfied in a condition of higher effective temperature, which would of course be more economical to maintain.

ACKNOWLEDGMENT

The authors acknowledge the valued services rendered by members of the staff of the Board of Governors of the Federal Reserve System, and its building manager, F. E. Spurney, in making the study possible, and for all their services in collecting and tabulating the data.

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[6] Loc. Cit. Note 2.
[7] Loc. Cit. Notes 2, 4, 5.

The authors also acknowledge the valuable assistance rendered by T. H. Urdahl and other members of the Washington, D. C., Chapter of the ASHVE in making arrangements for the investigation.

DISCUSSION

DR. C. A. MILLS (WRITTEN): I wish to caution Mr. Houghten and his collaborators, in this study of summer cooling requirements, against considering man a static and fixed being. Man's internal heat production is highly responsive to changing conditions of external warmth and the ease with which body heat can be lost. Pittsburgh people may indeed show a comfort zone within the same effective temperature range as do residents of Texas, but only after they have been forced to adapt themselves, for 10 days or more, to external warmth fully as severe as that of a Texas summer.

In all their studies on summer cooling requirements for different metropolitan areas, Mr. Houghten and his collaborators have failed to take into account one important factor, whether or not the test subjects had been exposed to the prevailing outside daytime heat. In the Pittsburgh study, when the summer comfort zone was found similar to that desired by Texas residents, the Pittsburgh test subjects were living and working in unconditioned quarters during unprecedented summer heat. And in their recent study reporting a comfort zone for San Antonio office workers the same as that for Pittsburgh residents, they again made the mistake of using as test subjects people who had been working steadily in cooled quarters from the onset of summer warmth. Such workers had been subjected to no depressive heat, particularly in a region where the nights are quite comfortable, and would thus be expected to maintain a high internal combustion level and be well able to stand the same cooling as Pittsburgh people.

In the report submitted today on Washington cooling requirements, the test subjects were apparently people who had been working for months in the conditioned building, although no mention is made of the duration of their employment in this building. Furthermore, no severe or prolonged heat afflicted Washington residents last summer preceding the study. It is not surprising, therefore, that the comfort zone for workers in cooled daytime quarters, with no severe night time warmth, should be found at a normal level. In the first Texas study, when previously unconditioned test subjects were used, the comfort zone was found to be distinctly higher.

It would seem essential that studies of this type be more thoroughly controlled from a physiologic standpoint, before authoritative statements are published under the Society's auspices, statements which may be misleading because all factors have not been sufficiently considered. Summer and winter levels of effective comfort temperatures can be obtained only with people not subjected to summer warmth. Texas residents can show the same comfort requirements as Pittsburgh people only when they are not exposed to outside daytime heat. Studies, such as those being made and reported to the Society, should deal separately with people already adapted to summer daytime cooling, as contrasted to those who have had to live in unconditioned quarters. Differences in summer cooling requirements depend essentially upon variations in body heat production and the necessity of dissipating this heat into the surrounding environment. And no longer can there be any reasonable doubt that internal combustion rate depends upon the ease with which body heat can be lost. We discussed this point rather fully in a recent number of the *American Journal of Hygiene* (May 1939). Only with a thorough appreciation of this fact can studies on summer cooling or winter heating requirements be intelligently carried out. Man is not a standardized

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being in his heat producing mechanism, but is highly responsive to the ease with which his waste heat can be dissipated.

T. H. URDAHL (WRITTEN): The temperature at which occupants are most comfortable is the first fundamental in air conditioning of buildings. What then, is more valuable than the exact knowledge of the temperature at which most people attain the greatest sensation of comfort? The very extensive task of obtaining this important fundamental which has been so carefully developed through the painstaking and accurate assembly of data and its analysis by the authors can scarcely be portrayed to the reader of the paper or the user of these data.

It is a natural human weakness of all of us to under-estimate the intangible. To illustrate, I was discussing some of the interesting facts produced in the paper with a neighbor while mowing our adjoining lawns the other evening. He is a prominent local physician and would have a natural interest in such matters. However, after some discussion he summed up his thoughts with the statement, "Well, we've been taking statistics with a grain of salt for some time. Today we have to take a whole spoonful." Few people realize the effort and expense that has to be put forth to obtain the few homely but essential facts which form the roots of what we are now accustomed to look upon as commonplace items contributing to our greater enjoyment of life.

A recent article in the *Business Week* tells of a pair of lady's hose exhibited at the World's Fair in New York and developed at a cost of $250,000.00. In developing the new material new opportunities are being developed for countless thousands through its effects in allied fields, of which air conditioning too, seems a very essential element. But, compare this amount with our entire research budget and the results obtained.

I only wish it were possible for each member to make an actual and visual comparison between the quantity and quality of fundamental work turned out per dollar by the Research Laboratory and others whose operating budgets are many times as great. Then, I am sure, each one would share with me a greater measure of personal pride in each project of which this paper is an outstanding example.

M. G. KERSHAW (WRITTEN): I have read this paper with interest, and note the reference to maintaining effective temperatures and that the belief is that the shock item should not be serious.

In operating air conditioning systems we have found that in the afternoon as the outside temperature climbs, it is necessary to raise the inside temperature so that when the occupants leave the building, the temperature difference is not too great, otherwise we receive complaints from shock. We have tentatively decided to try 75 F inside when it is 75 F outside and to have 83 F inside when it is 95 F outside.

71 F EFFECTIVE COMFORT LINE

D. B.	W. B.	D. P.	R. H. Per Cent
75	66	61.3	63
76	64	57.4	53
77	62	53.0	43
78	61	50.0	38
79	60	47.0	30
80	58	40.0	25
81	56	33.0	18
82	54	23.0	11
83	52	8.0	6

74 F EFFECTIVE COMFORT LINE

D. B.	W. B.	D. P.	R. H. Per Cent
75	73	72.4	90
76	72	70.4	82
77	70	67.0	70
78	69.5	66.0	68
79	68	63.8	58
80	67	60.2	50
81	66	58.0	45
82	64	53.7	38
83	63	50.8	32

This change will be progressive. If we decide to maintain 71 F effective comfort line we figure the conditions in the tabulation on opposite page will be required.

Most systems are designed to maintain 80 F inside with 50 per cent relative humidity when it is 95 F outside. This gives an effective temperature of 74 F. If we desire to maintain a 74 F effective comfort line the conditions outlined in the above tabulation will be required.

As you can see from the tabulations constant resetting of the dew-point will be necessary and in the case of the 71 F comfort line is not practical. Apparently some arbitrary dew-point will have to be picked, so that with the automatic resetting of the dry-bulb, we will have a sliding scale effective comfort line.

We are wondering if the work to date gives enough information so that one could make suggestions as to the proper sliding scale to meet practical operating conditions. We note from the curves that an 80 per cent comfort is indicated for a wide range of from 69½ to 73½.

DR. M. B. FERDERBER: The results of the comfort study in the cities, as explained, bring up a very interesting point in connection with the air conditioned operating room in Pittsburgh. When the room was completed for use, the question of proper temperatures was raised, and the idea was advanced that those conditions should be any given number of degrees cooler than the outside environment. Mr. Houghten felt that the comfort studies just described, some of which he had known of before, might be an excellent index of the proper conditions which should prevail in the operating rooms. Accordingly, this opinion was followed based on his previous information and the conditions of the operating room closely approximated those of the comfort studies. To the satisfaction of all concerned, it was remarkable how well these same figures given in this paper approached the conditions in the hospital room. This would seem to break down the old concept that the operating room should be so many degrees cooler than the outside temperature and again proves the wisdom of the Society's method of gathering and expressing data. Although the operating room study revealed that some of the upper and lower limits of effective temperature were not exactly those of the comfort study, it is quite definite that the average graph would resemble one of these which Mr. Houghten has just displayed. In my opinion this excellent presentation should serve as a yardstick for measuring sensations of comfort in large groups.

C. TASKER: Director Houghten has told you that this paper not only records the results of the studies made in the Federal Reserve Building at Washington during

the summer of 1938, but summarizes the results of the many studies made during the past 3 years, on summer cooling requirements.

These studies were made under the Technical Advisory Committee on Sensations of Comfort, and as chairman of that Committee I should like to take this opportunity to express, not only on behalf of the members of that Committee but on behalf of the Committee on Research as a whole, our very deep sense of appreciation to all those organizations and individuals who have had a part in these studies. I need not remind the members of this Society that with the budget at the disposal of the Research Committee it would have been absolutely impossible for us to have financed this work ourselves. We have contributed from our funds some small sums of money to assist certain of the Laboratory studies, while Director Houghten and some of his staff have spent a certain amount of time advising on the collection of the data and analyzing it and preparing it for presentation to the Society. The main cost, however, has been borne by those institutions and organizations cooperating with us on the work and to them we offer our grateful thanks.

While this phase of the subject has now been carefully studied and concordant and satisfactory findings made, there remains much to be done. Maintaining the same effective temperatures, we are very anxious to investigate the effect of lower humidities and higher dry-bulb temperatures than were used in these studies. In this phase of our studies we hope to secure the cooperation of the *American Gas Association,* but we first need to find buildings suitably equipped and then persuade the executives of the value of such studies.

Another very important question is that of short term occupancy. In all the studies made to date and summarized in this paper the occupants remained in the conditioned space for 1 to 2 hours and few, if any, reactions were reported until after the subjects had been exposed to the environmental conditions for 20 min. Short term occupants in banks, department stores, etc., may have reactions differing slightly or widely form those of the office workers who in nearly all cases were seated and engaged in their normal occupations. This phase of the subject needs careful study and presents many difficulties in the collection and interpretation of data which were not met with in our studies to date. It needs most of all the helpful cooperation of those in a position to make such studies possible.

No. 1137

STUDY OF SUMMER COOLING IN THE RESEARCH RESIDENCE FOR THE SUMMER OF 1938

By A. P. KRATZ,* S. KONZO,** M. K. FAHNESTOCK,*** AND E. L. BRODERICK,† (*MEMBERS*), URBANA, ILL.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the *National Warm Air Heating and Air Conditioning Association* and the University of Illinois.

DURING the summer of 1937 an investigation [1] was undertaken to determine the temperature and humidity conditions that could be maintained in the Research Residence with a mechanical condensing unit having a capacity approximately 40 per cent less than the 30,000 Btu per hour used during the summers of 1932 to 1936 inclusive.

These tests with a small capacity, mechanical condensing unit were extended in 1938 and, in addition, were supplemented by tests made with the unit operated at a capacity of 30,000 Btu per hour. In the latter case, cooled air was delivered only to the rooms on the second story.

DESCRIPTION OF COOLING EQUIPMENT

The Research Residence together with the forced-air heating system has been described in previous papers.[1, 2] In every respect the state of the Residence during 1938 was the same as that during the preceding summer. The

* Research Professor, Engineering Experiment Station, University of Illinois.
** Special Research Assistant Professor, Engineering Experiment Station, University of Illinois.
*** Research Assistant Professor, Engineering Experiment Station, University of Illinois.
† Research Assistant, Engineering Experiment Station, University of Illinois.

[1] ASHVE RESEARCH REPORT No. 1095—Study of Summer Cooling in the Research Residence Using a Small Capacity Mechanical Condensing Unit, by A. P. Kratz, S. Konzo, M. K. Fahnestock and E. L. Broderick. (ASHVE TRANSACTIONS, Vol. 44, 1938, p. 439.)

[2] Investigation of Summer Cooling in the Warm-Air Heating Research Residence, by A. P. Kratz, M. K. Fahnestock, and S. Konzo. (University of Illinois, *Engineering Experiment Station Bulletin* No. 290, January, 1937.)

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Mackinac Island, Mich., July, 1939.

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arrangement of the forced-air duct system and fan was also the same for the two summers and was such that the cooled air could be delivered entirely through

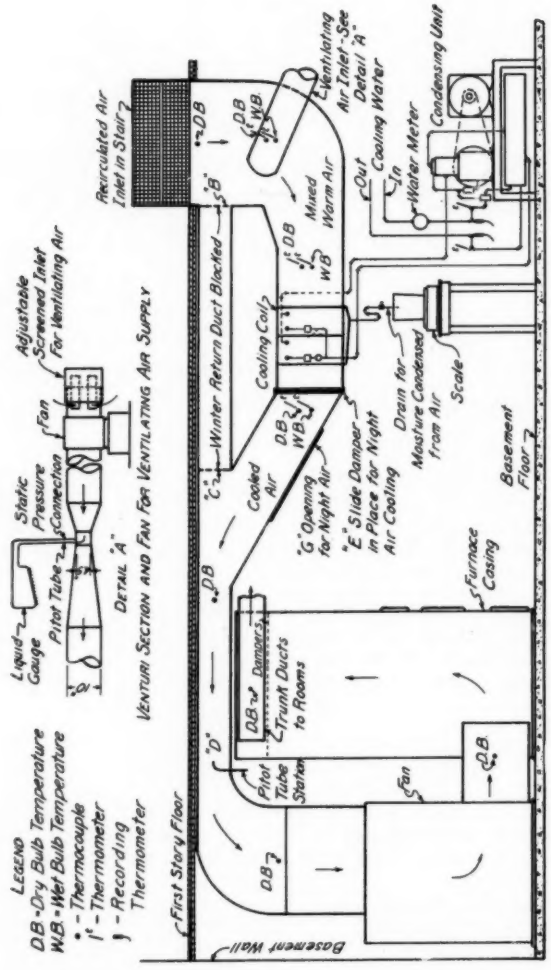

FIG. 1. DIAGRAM OF COOLING PLANT WITH MECHANICAL REFRIGERATION

the registers on the first story, entirely through the registers on the second story, or through the registers on both stories.

The arrangement of the cooling plant, shown in Fig. 1, was practically the

same as that fully described in a previous publication[3] except that two evaporator units were installed, and provision was made to use either one unit alone or both units in parallel. The maximum capacity of the condensing unit was approximately 30,000 Btu per hour with both evaporator units in use and the compressor running at 429 rpm. However, for the first part of the season (*Series* 1-38, 2-38) the capacity of the condensing unit was reduced to 18,000 Btu per hour by changing a pulley on the motor, which lowered the compressor

TABLE 1—FAN DATA

(*Calculations based on air density of* 0.075 *lb per cubic foot*)

SERIES NO.	CAPACITY OF UNIT BTU PER HOUR	AIR DISTRIBUTION	AIR VOLUME, CFM	AIR WEIGHT, LB PER HOUR	AIR CHANGES[a] PER HOUR
COOLING WITH OUTDOOR AIR AT NIGHT					
1–38 and 2–38	....	Both Stories	1560	....	6.6
4–38	....	Both Stories	1890	....	8.0
AIR RECIRCULATION DURING DAY INCLUDING VENTILATING AIR					
1–38	18,000	Both Stories	1020[b]	4590	4.3
2–38	18,000	2nd Story only	880[b]	3960	7.4[c]
3–38 and 4–38	30,000	2nd Story only	1040[b]	4680	8.7[c]
VENTILATING AIR					
For all series	....	....	228	1024	1.0

[a] Outdoor air admitted for ventilation amounting to one air change per hour included in these values.
[b] Quantity of air through wet coil.
[c] Based on second story only.

speed, and by closing a valve in the refrigerant line, which cut off one of the two evaporators.

A study of heat flow through the exposed west and north walls of the northwest bedroom of the Research Residence was begun in 1937.[4] These wall sections consisted of lap siding, building paper, sheathing, studding space, wood lath and plaster with rough sand finish. This study was extended in 1938 to include tests on an insulated wall section. For this purpose the west wall studding space, which was 5⅝ in. deep, was filled with mineral wool. The calculated coefficients of heat transmission for the uninsulated and insulated walls were respectively, 0.25 and 0.042 Btu per square foot per hour per degree Fahrenheit at a wind velocity of 15 mph. The north wall was not insulated. The thermocouples and the heat flow meters which were used in 1937 were also used in 1938 tests.

[3] Loc. Cit. See Note 1.
[4] Ibid.

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TABLE 2—COMPLETE DATA AND RESULTS FOR TESTS—1938

DATE 1938	TEST No.	SERIES No.	WEATHER DATA							INDOOR AIR CONDITIONS				TEMPERATURE DIFFERENCE		NIGHT AIR COOLING PREVIOUS TO TEST PERIOD	
			Degree-Hours		Total Sunshine in Hours and Minutes	Outdoor Air				At End of Night Cooling		Avg During Period Requiring Cooling		Outdoor Max. to Avg Indoor, F	Avg Outdoor to Avg Indoor, F	Duration, Hours	Power, Kwhr
			Above 80 F	Above 85 F		Maximum Temp. During Test, F	Minimum Temp. Previous to Test, F	Avg During Period Requiring Cooling Temp., F	Avg During Period Requiring Cooling Hum., Per Cent	Temp., F	Hum., Per Cent	Temp., F	Hum., Per Cent				
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
6–21	1	1–38	39.7	3.0	15:06	87	62	82.7	39.6	71.8	60.9	76.8	51.3	10.2	5.9	11.25	5.2
6–22	2	1–38	20.9	0	7:10	85	64	79.4	52.3	72.7	63.7	77.0	59.4	8.0	2.4	10.67	5.1
6–23	3	1–38	75.8	29.3	12:51	92	64	86.9	51.4	73.7	71.9	78.4	60.6	13.6	8.5	10.50	5.0
6–24	4	1–38	7.4	0	3:10	82	72	77.9	82.5	77.6	80.9	77.2	66.4	4.8	0.7	9.83	4.6
6–25	5	1–38	52.3	20.0	9:35	90	69	85.9	59.2	75.2	83.5	78.1	66.7	11.9	7.8	8.33	3.8
7–1	6	1–38	22.0	2.0	4:43	90	70	82.4	73.3	75.8	82.8	77.2	72.5	12.8	5.2	0	0
7–2	7	1–38	1.0	0	2:25	80	68	73.3	79.8	74.9	85.7	76.6	76.2	3.4	−3.3	12.92	6.3
7–3	8	1–38	5.0	0	9:40	82	69	80.3	62.4	73.8	80.4	77.1	67.1	4.9	3.2	17.25	0
7–4	9	1–38	3.0	0	12:35	82	63	79.0	54.7	72.0	61.9	77.0	58.3	5.0	2.0	11.67	5.6
7–5	10	1–38	13.4	0	13:34	82	63	80.3	49.8	71.5	65.3	77.1	54.9	4.9	3.2	11.67	5.4
7–6	11	1–38	51.2	11.4	14:49	87	65	82.9	59.0	72.4	66.7	77.5	56.3	9.5	5.4	11.50	5.6
7–7	12	1–38	71.7	29.6	12:25	90	68	87.2	54.5	75.3	71.4	78.8	61.0	11.3	8.5	9.17	4.4
7–8	13	1–38	45.1	6.9	10:43	86	68	83.2	61.0	75.3	72.9	77.8	61.1	8.2	5.4	12.50	6.3
7–9	14	1–38	60.4	13.8	14:56	90	66	85.3	49.8	75.3	70.2	77.8	57.0	12.2	7.5	9.25	4.5
7–10	15	1–38	83.2	32.2	12:39	90	68	85.8	53.7	75.1	68.1	78.5	57.2	11.5	7.3	10.75	5.1
7–11	16	1–38	93.7	38.4	11:57	96	74	85.6	56.3	77.6	72.2	79.4	57.7	16.6	6.2	8.42	4.0
7–12	17	1–38	79.6	34.6	10:50	92	70	85.5	57.8	76.9	70.5	79.5	61.3	12.5	6.0	9.25	4.3
7–13	18	1–38	38.4	2.6	6:20	86	70	79.7	67.3	75.8	65.1	77.4	59.6	8.6	2.3	10.50	5.1
7–14	19	1–38	38.6	0	14:41	85	68	81.6	45.8	75.4	57.5	77.3	51.1	7.7	4.3	0	0
7–15	20	1–38	9.1	0	12:52	84	62	80.4	50.2	72.1	62.1	77.5	53.8	7.5	2.9	12.00	5.9
7–16	21	1–38	52.1	12.4	11:55	88	64	83.4	54.9	71.8	66.1	77.7	56.9	10.3	5.7	12.25	6.1
7–19	22	1–38	23.1	0	14:35	86	64	82.1	51.0	71.2	66.9	77.1	55.5	8.9	5.0	11.92	5.8
7–20	23	1–38	32.5	7.4	10:02	87	66	83.9	57.4	72.6	68.5	77.2	62.1	9.8	6.7	11.75	5.6
7–23	24	1–38	16.3	0	14:50	85	62	80.6	50.9	71.5	74.8	77.3	59.4	7.7	3.3	13.00	6.3
7–24	25	1–38	35.0	0.7	14:36	88	62	82.1	59.3	70.9	68.5	77.3	59.8	10.7	4.8	12.17	5.8
7–25	26	1–38	70.5	24.2	12:51	91	62	85.2	59.2	73.1	68.5	78.2	61.0	12.8	7.0	10.92	5.1
7–26	27	1–38	27.9	6.6	8:15	89	69	82.4	61.7	75.2	72.1	77.4	63.8	11.6	5.0	9.00	4.2
7–27	28	2–38	13.4	0.3	5:25	87	68	79.2	76.7	73.8	83.1	77.2	74.0	9.8	2.0	16.00	7.6
7–28	29	2–38	3.9	0	13:08	87	67	82.9	56.9	73.6	82.5	77.5	62.2	9.5	5.4	13.00	6.0
7–29	30	2–38	10.0	0	12:04	84	65	80.8	53.4	73.3	73.1	77.6	58.9	6.4	3.2	11.25	5.3
7–30	31	2–38	20.6	0	12:46	83	64	81.0	69.0	72.5	70.4	77.4	64.7	5.6	3.6	12.00	5.8
7–31	32	2–38	2.6	0	6:45	81	69	80.1	69.9	75.3	80.0	76.5	65.9	4.5	3.6	11.33	5.4
8–1	33	2–38	1.0	0	8:20	85	64	79.4	71.4	72.4	75.3	76.4	70.1	8.6	3.0	12.50	6.0
8–2	34	2–38	48.9	10.4	14:18	88	67	84.4	64.9	73.6	81.4	77.0	64.9	11.0	7.4	11.50	5.4
8–15	43	4–38	24.3	0	3:15	83	71	81.8	73.1	No Night	Cooling	78.0	63.4	5.0	3.8	0	0
8–16	44	4–38	19.1	0	4:05	84	70	81.1	72.7	75.4	84.8	78.4	76.5	5.6	2.7	9.50	6.0
8–20	45	4–38	49.0	11.8	7:55	89	66	81.7	68.3	72.0	72.9	78.0	71.0	11.0	3.7	12.42	7.4
8–23	46	4–38	77.4	34.0	11:17	95	71	86.4	63.8	75.2	79.8	77.8	62.2	17.2	8.6	12.00	7.6

TABLE 2 (*Continued*)—COMPLETE DATA AND RESULTS FOR TESTS—1938

OPERATION OF THE FAN				OVER-ALL TEST PERIOD			COMPRESSOR OPERATION					TOTAL MOISTURE CONDENSED, LB	HEAT ABSORBED BY COOLING COIL DURING OVER-ALL TEST PERIOD, BTU			RATIO MOISTURE TO TOTAL LOAD
Recirculation Previous to Test Period		Recirculation During Test Period		Start of Cooling with Condensing Unit to Start of Night Air Cooling			Time in Hours	Power Input, Kwhr	Condenser Water							
									Quantity in Gallons	Temperature						
Duration, Hours	Power, Kwhr	Duration, Hours	Power, Kwhr	Start	End	Duration, Hours				Inlet, F	Rise, F		Load Due to Moisture	Sensible	Total	
19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35
7.33	3.0	6.00	2.7	2:20p	8:20p	6.00	5.70	7.4	293.6	58.3	40.4	12.65	14,250	84,550	98,800	0.144
8.25	3.6	5.25	2.3	3:15p	8:30p	5.25	2.72	3.5	158.0	58.2	37.4	9.73	11,050	38,450	49,500	0.223
5.08	2.0	9.08	4.2	12:05p	9:10p	9.08	8.92	11.4	532.0	58.1	38.5	48.92	55,100	115,400	170,500	0.323
0	0	15.67	7.1	7:00a	10:40p	15.67	14.00	18.2	821.0	58.3	39.2	102.59	114,650	153,350	268,000	0.428
2.25	1.0	9.00	4.3	9:15a	6:15p	9.00	9.00	11.7	519.0	58.5	40.5	64.43	71,800	103,200	175,000	0.410
4.33	1.8	2.75	1.4	3:20p	6:05p	2.75	2.58	3.6	150.5	58.7	40.4	19.79	22,890	27,710	50,500	0.454
3.00	1.2	3.75	1.8	10:00a	1:45p	3.75	1.05	1.3	63.3	58.9	40.5	8.84	9,840	11,490	21,330	0.461
5.75	2.5	6.58	3.1	12:45p	7:20p	6.58	4.58	5.9	259.0	59.4	39.8	27.09	30,200	55,700	85,900	0.352
9.00	4.1	3.33	1.5	4:00p	7:20p	3.33	2.65	3.7	142.3	59.6	39.8	9.16	11,280	25,920	47,200	0.239
8.50	3.9	4.00	1.6	3:30p	7:30p	4.00	3.82	4.9	199.6	59.1	40.2	10.90	12,510	54,390	66,900	0.187
6.25	2.9	8.58	3.8	1:15p	9:50p	8.58	8.50	11.1	462.9	59.1	40.3	40.39	45,500	109,900	155,400	0.293
2.50	1.2	9.00	4.1	9:30a	6:30p	9.00	9.00	11.7	510.2	59.0	40.4	52.20	58,950	112,750	171,700	0.343
2.50	1.0	12.25	5.7	9:30a	9:45p	12.25	12.08	15.7	676.6	59.6	40.3	70.44	82,600	144,559	227,159	0.363
2.25	1.0	11.00	5.2	9:15a	8:15p	11.00	11.00	14.3	597.5	59.1	40.2	47.75	53,800	146,275	200,075	0.269
3.00	1.5	12.58	6.0	10:00a	10:35p	12.58	12.33	15.9	683.0	59.4	40.0	55.57	62,750	164,720	227,470	0.276
0	0	14.75	7.0	7:00a	9:45p	14.75	14.50	18.8	818.2	59.0	40.4	76.89	88,250	187,096	275,346	0.320
0.58	0.3	12.92	6.1	7:35a	8:30p	12.92	12.67	16.6	730.0	59.3	39.8	78.24	88,900	153,200	242,100	0.367
3.33	1.5	20.67	9.8	10:20a	7:00a	20.67	18.03	23.1	1014.0	59.8	39.4	102.00	114,200	218,700	332,900	0.313
1.50	0.5	10.50	4.9	8:30a	7:00p	10.50	9.85	12.7	516.0	59.4	39.6	25.88	29,900	140,100	170,000	0.176
9.25	4.0	2.50	1.3	4:15p	6:45p	2.50	2.25	2.8	117.4	60.0	39.2	5.61	6,580	31,750	38,330	0.172
7.00	3.0	6.25	2.9	2:00p	8:15p	6.25	6.00	7.5	334.4	58.9	40.3	24.87	28,300	83,580	111,880	0.253
8.00	3.5	4.25	2.0	3:00p	7:15p	4.25	4.00	5.3	212.2	59.9	39.3	14.28	16,190	53,260	69,450	0.233
5.25	2.3	3.50	1.7	12:15p	3:45p	3.50	3.18	4.3	176.1	59.6	39.5	15.46	17,220	40,780	58,000	0.297
9.33	4.1	2.67	1.2	4:20p	7:00p	2.67	2.42	3.2	130.4	60.0	39.2	8.06	9,110	33,470	42,580	0.214
8.50	3.8	4.42	2.1	3:40p	8:05p	4.42	4.20	5.4	234.1	60.1	39.0	21.20	23,700	51,950	75,650	0.313
4.67	2.1	10.33	4.8	11:40a	10:00p	10.33	10.33	13.6	586.0	59.4	39.7	59.85	68,200	125,400	193,600	0.352
2.05	0.9	5.95	2.6	9:03a	3:00p	5.95	5.78	7.2	326.0	59.1	39.9	35.72	40,720	67,580	108,300	0.376
4.67	2.1	6.33	2.9	11:40a	6:00p	6.33	4.17	5.6	244.7	60.2	38.9	32.07	36,080	43,200	79,300	0.455
5.83	2.6	6.92	3.0	12:50p	7:45p	6.92	6.50	8.6	371.0	60.0	39.2	35.13	40,100	81,000	121,100	0.331
7.00	3.0	5.00	2.2	2:00p	7:00p	5.00	4.50	5.8	240.9	59.7	39.5	16.50	19,300	59,920	79,220	0.244
7.00	3.1	5.67	2.5	2:00p	7:40p	5.67	5.17	6.7	290.5	59.9	39.2	32.10	36,200	58,700	94,900	0.382
8.00	3.5	3.50	1.6	3:00p	6:30p	3.50	3.50	4.4	204.8	61.0	38.2	23.63	26,800	38,400	65,200	0.411
10.17	4.5	2.33	1.1	5:10p	7:30p	2.33	1.42	1.9	82.8	60.8	38.4	8.85	10,050	16,450	26,500	0.379
5.15	2.7	8.85	3.6	12:09p	9:00p	8.85	8.85	11.7	520.0	60.4	38.8	60.68	68,600	99,300	167,900	0.409
5.00	2.9	9.50	5.9	12:00	9:30p	9.50	5.18	12.0	590.0	59.7	38.2	59.08	65,400	122,100	187,500	0.349
7.22	4.1	8.28	4.9	2:13p	10:30p	8.28	2.02	4.8	239.5	....	....	16.40	....	....	....	....
9.43	5.7	3.07	1.8	4:51p	7:55p	3.07	0.75	2.0	83.9	....	....	2.49	....	....	....	....
5.42	3.1	9.35	5.7	12:25p	9:46p	9.35	6.08	15.0	694.0	59.3	37.9	57.86	64,050	155,050	219,100	0.292

METHOD OF CONDUCTING TESTS

Continuous records were made during the summer of air temperatures in the various parts of the house and in the different sections of the duct system. During the periods of operation, observations were made of the weight of water entering and leaving both the condensing unit and the house, the head and suction pressures, and the electrical inputs to the compressor and fan motors.

During all of the tests the windows in the attic remained open and those on the first story remained closed. The recirculated air was returned through the return air grille located in the stair risers in the first story hall. For the purpose of cooling with outdoor air at night, the attic door and 11 windows on the second story were opened in the evening when the effective temperature outdoors became equal to the effective temperature indoors. The dampers were set and the basement door opened, so that the fan delivered outdoor air through the duct system to both stories until 7:00 a.m. The windows and doors were closed at 7:00 a.m. and the dampers were changed so that the fan started delivering recirculated air and the outdoor air admitted for ventilation. The fan was allowed to run continuously during the day and night. The quantity of air delivered by the fan for each test series and for cooling with outdoor air at night is shown in Table 1.

The Residence was operated strictly on the schedules described in connection with the following discussion of the different test series:

Simultaneous Cooling on Both Stories (*Series* 1-38)

After 7:00 a.m. the air was recirculated on both stories. When the average temperature on the second story rose to about 77 F, the condensing unit, operating at a capacity of 18,000 Btu per hour, was started and cooled air was distributed to the rooms on both stories.

The method of operation used in this series was the same as that for *Series* 1 of 1937 except that the cooling plant was started at a lower indoor temperature. In extremely mild weather the cooling plant operated intermittently and maintained an indoor temperature of 77 F. However, the heat absorbing capacity of the unit was so small that on most days when cooling was required, a slight rise in indoor temperature occurred as the test progressed, and the plant operated continuously during the whole period.

Cooling on Second Story with Small Capacity Unit (*Series* 2-38)

The air was distributed only to the second story rooms after 7:00 a.m. When the average temperature on the second story rose to about 77 F, the condensing unit, operating at a capacity of 18,000 Btu per hour, was started and cooled air was distributed only to the rooms on the second story.

Cooling on Second Story to Determine Rate of Reduction of Temperature on Second Story (*Series* 3-38)

The air was distributed only to the first story rooms after 7:00 a.m. and the condensing unit, operating at a capacity of 30,000 Btu per hour, operated intermittently maintaining a dry-bulb temperature of approximately 79 F. At

some time during the early evening, when the temperature of the air on the second story had risen to a value exceeding 82 F, the dampers were changed so that all of the cooled air was distributed to the second story rooms alone. The cooling plant operated continuously until the average temperature on the second story was reduced to 78.5 F.

Cooling on Second Story with Large Capacity Unit (*Series* 4-38)

The method of operation was the same as that for *Series* 2-38 except that the condensing unit was operated at a capacity of 30,000 Btu per hour. The unit was controlled to maintain a temperature of approximately 78 F on the second story.

RESULTS OF TESTS

Simultaneous Cooling on Both Stories (*Series* 1-38)

The results obtained from this series of tests were practically the same as those obtained from the work done during the preceding summer. General data and results from all of the tests are shown in Table 2.

The operating characteristics of the cooling plant can best be illustrated by the results obtained on a typical day. For this purpose the test made on June 23, 1938, was selected as typical of the operation obtained when the rooms on both stories were cooled (*Series* 1-38), and the results are shown in the first column of Table 3 and in Fig. 2. On this test the second story windows had been open during the preceding night from 8:30 p.m. to 7:00 a.m., and outdoor air had been circulated by means of the basement fan. The windows were closed at 7:00 a.m. and recirculation of the air in the house was begun. The condensing unit was started at 12:05 p.m.

The tests made during the summer of 1937 had shown that with high humidity conditions at the start of a test, a starting temperature of about 78 F resulted in an indoor effective temperature higher than the 75 deg representing the upper border line of comfort. In 1938, however, the indoor effective temperature at the start of the tests was not allowed to exceed 75 deg. This was accomplished by starting the condensing unit at the lower indoor temperature of about 77 F, as shown in Fig. 2. The condensing unit operated continuously from 12.05 p.m. to 9:00 p.m. maintaining the dry-bulb temperature between 77.4 F and 79.2 F. Three hours after the condensing unit started the relative humidity stabilized at about 59 per cent and the effective temperature did not exceed 75 deg. The indoor conditions during this period could be considered as entirely satisfactory.

It is evident from the curve of indoor dry-bulb temperature in Fig. 2 that the condensing unit was insufficient in capacity to maintain a constant temperature. If the temperature of the outdoor air had been materially higher, the rise in the indoor temperature would have been greater and the maximum indoor effective temperature would probably have attained a value above that represented by the upper limit of the comfort zone. Since the season was comparatively cool, no test data under severe weather conditions were obtainable.

In the study made in 1937 a relationship was shown between the maximum indoor air temperature obtained in the house and the daily median outdoor

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TABLE 3—TYPICAL OPERATING DATA AND RESULTS

1. Date	6-23-38	8-2-38	8-23-38
2. Test No.	3	34	46
3. Series	1-38	2-38	4-38
4. Time of Day	2:10 p.m.	2:20 p.m.	4:40 p.m.
5. Outdoor Air			
Dry-Bulb, F	90.5	87.0	90.9
Wet-Bulb, F	73.7	75.2	77.7
Dew-Point, F	66.5	70.7	72.9
Relative Humidity, per cent	44.3	57.5	54.5
Specific Humidity, grains per pound dry air	97.5	112.7	121.9
6. Indoor Air, average breathing level temperature			
1st Story, F	77.7	76.5	77.6
2nd Story, F	77.8	76.8	77.8
7. Indoor Air, average at breathing level			
Dry-Bulb, F	77.8	76.7	77.7
Wet-Bulb, F	68.6	68.6	67.1
Dew-Point, F	64.2	64.7	61.9
Relative Humidity, per cent	62.2	66.0	57.6
Specific Humidity, grains per pound dry air	89.8	91.4	82.3
8. Indoor Air, entering return grille			
Dry-Bulb, F	77.4	76.7	77.1
Wet-Bulb, F	67.9	68.4	66.4
Dew-Point, F	63.2	64.5	60.9
Relative Humidity, per cent	61.0	65.6	56.5
Specific Humidity, grains per pound dry air	86.6	90.6	79.6
9. Ventilating Air			
Dry-Bulb, F	88.1	85.1	88.1
Wet-Bulb, F	72.4	74.8	76.0
Dew-Point, F	65.5	70.7	71.3
Relative Humidity, per cent	46.1	61.2	56.5
Specific Humidity, grains per pound dry air	94.2	113.1	115.5
10. Mixed Air, entering cooling coil			
Dry-Bulb, F	79.5	78.5	79.4
Wet-Bulb, F	68.9	69.8	68.6
Dew-Point, F	63.9	65.8	63.3
Relative Humidity, per cent	58.0	64.3	57.0
Specific Humidity, grains per pound dry air	88.7	95.1	86.9
11. Mixed Air, leaving cooling coil			
Dry-Bulb, F	67.6	66.6	58.8
Wet-Bulb, F	63.1	63.4	57.3
Dew-Point, F	60.6	61.7	56.3
Relative Humidity, per cent	78.0	83.9	91.8
Specific Humidity, grains per pound dry air	78.9	81.9	67.6
12. Air Temperature drop through cooling coil, F	12.0	11.9	20.6
13. Temperature cooled air leaving registers, average, F	71.0	70.6	68.8
14. Air Temperature rise in ducts and casing, F	3.5	4.0	10.0
15. Basement Air Temperature at breathing level, F	72.8	73.1	73.7
16. Cooling Coil			
Gross Face Area, sq ft	2.07	2.07	2.07
Net Free Area, sq ft	1.02	1.02	1.02
17. Air Velocity through coil			
Gross Face, fpm	493	425	503
Free Area, fpm	1,000	863	1,020
18. Moisture condensed from air, pounds per hour	5.88	7.04	10.59
19. Heat given up by air; total Btu per hour	19,720	19,250	34,900
Heat due to moisture in air			
Btu per hour	6,620	7,950	11,720
Per cent of total heat	33.6	41.3	33.6
Sensible heat			
Btu per hour	13,100	11,300	23,180
Per cent of total heat	66.4	58.7	66.4
20. Water Temperature in condenser			
Inlet, F	58.2	60.4	59.3
Outlet, F	95.0	99.2	97.9
Rise, F	36.8	38.8	38.6
21. Water Temperature in house			
Inlet, F	58.0	60.4	59.3
Outlet, F	95.2	98.8	97.8
Rise, F	37.2	38.4	38.5
22. Quantity of cooling water			
Pounds per hour	514	485	940
Gallons per hour	61.7	58.2	112.9
23. Heat absorbed by cooling water, Btu per hour			
Through condenser	18,920	18,820	36,280
Through house	19,120	18,625	36,190
24. Refrigerant pressures, gage			
Suction, pound per sq in	38	40	36
Discharge, pound per sq in	125	125	130
25. Nominal Rating of condensing unit with 140-lb head pressure, 55 F return refrigerant, 59 F condensing water and 74 F ambient air, Btu per hour	16,300	16,300	31,150
26. Compressor Speed, rpm	229	229	429
27. Compressor Motor Data			
Size, hp	2	2	2
Measured power rate, kw	1.28	1.32	2.47
28. Fan Speed, rpm	444	444	526
29. Fan Motor Data			
Size hp	0.75	0.75	0.75
Measured power rate, kw	0.43	0.44	0.60

temperature, or the average of the daily maximum and minimum outdoor temperatures. The results obtained have been reproduced and are shown as broken-line curves in Fig. 3. The maximum temperatures that were obtained in the house, when cooling was accomplished only by the use of outdoor air at night, are shown as the upper curve. For a day on which the median temperature did not exceed 75.5 F the indoor maximum temperature was 79 F and conditions were reasonably satisfactory if the relative humidity was not too

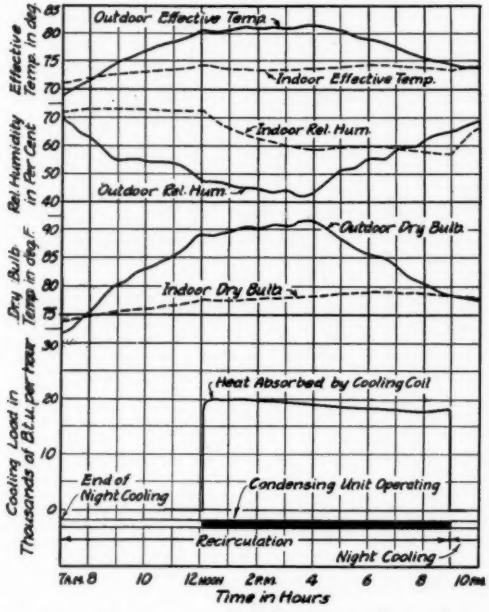

*FIG. 2. GRAPHIC LOG OF AIR TEMPERATURES, RELATIVE HUMIDITIES AND COOLING LOAD. TEST NO. 2. *Series* 1-38. JUNE 23, 1938. CAPACITY 18,000 BTU PER HOUR

high. The lower broken-line curve, which gives the maximum indoor air temperatures obtained when the small capacity condensing unit was operated, indicates that on days when the daily median temperature was 83 F or less the maximum indoor air temperature did not exceed 80 F.

The results obtained from tests made during 1938, shown as circles in Fig. 3, confirmed the results obtained in 1937, except that the starting of the condensing unit at a lower indoor temperature resulted in a lower maximum temperature on mild days. The lower starting temperature also required that the cooling plant be operated on days when cooling with outdoor air at night

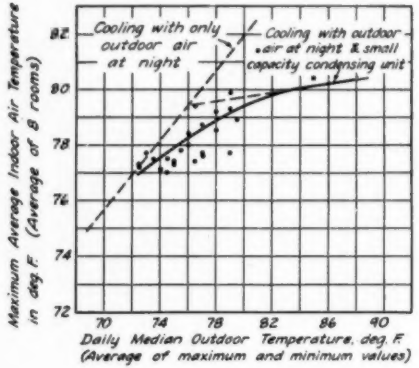

FIG. 3. VARIATION OF DAILY MAXIMUM INDOOR TEMPERATURE WITH OUTDOOR MEDIAN TEMPERATURE. RESEARCH RESIDENCE DATA. SUMMER OF 1938. (*Series* 1-38)

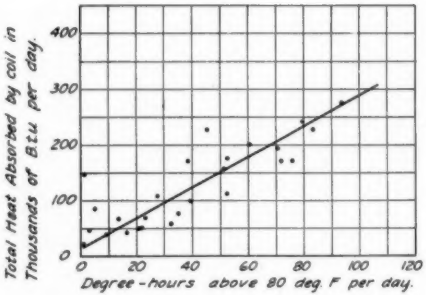

FIG. 4. HEAT ABSORBED BY COOLING COIL PER DAY. RESEARCH RESIDENCE DATA. (*Series* 1-38)

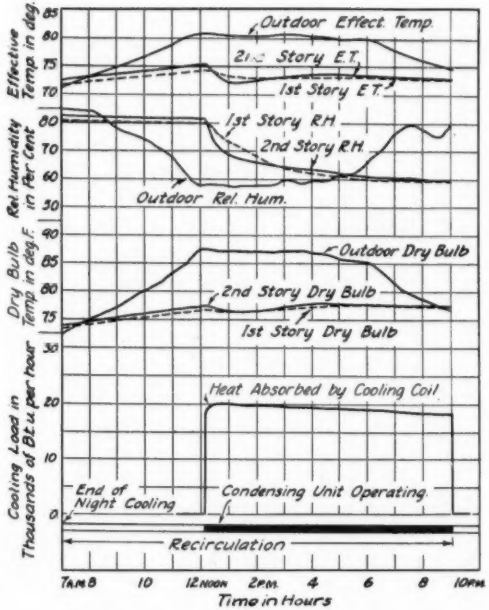

FIG. 5. GRAPHIC LOG OF AIR TEMPERATURES, RELATIVE HUMIDITIES AND COOLING LOAD. TEST NO. 34. *Series* 2-38. AUGUST 2, 1938. CAPACITY 18,000 BTU PER HOUR. RECIRCULATION ON SECOND STORY ONLY

alone would have given fairly satisfactory indoor conditions. The probable values of the maximum indoor air temperature, for operating conditions as maintained in 1938, are shown by the solid line in Fig. 3 which coincides with the curve for 1937 in the upper range. Since the daily median temperature of 83 F was exceeded only on three days during the cooling season of 1938, the capacity of the condensing unit was sufficient to maintain satisfactory conditions for practically the entire season. However, the analysis presented in a previous paper [5] had indicated that for an average season the small capacity condensing unit would not have maintained satisfactory conditions on 10 days during the season. This number would undoubtedly be exceeded in a severe season.

The total cooling load for the day, expressed as the total Btu absorbed from the air in the total amount of time during which the machine was actually in operation is shown in Fig. 4, plotted against the degree-hours per day. The base value of 80 F was selected in this case rather than that of 85 F which has heretofore been used, because it was found that when the machine was started at an indoor temperature of 77 F it was operated on many days on which the maximum outdoor temperature was less than 85 F. Considerable deviations of the points from the mean curve are to be expected since wind and sun effects, which influence the cooling load, are not directly dependent upon the degree-hours. Also the amount of cooling with air from outdoors at night is indirectly reflected in the cooling load required for a given number of degree-hours the next day. The curve shown in Fig. 4 can be used to estimate the seasonal amount of heat absorption required under similar conditions of plant operation for any season for which the number of degree-hours above 80 F corresponding to each day in the season is known.

Cooling on Second Story with Small Capacity Unit (*Series* 2-38)

The test made on August 2, 1938, was selected as typical of the operation obtained when the small capacity condensing unit was used and cooled air was delivered only to the rooms on the second story. The results are shown in the second column of Table 3 and in Fig. 5. Outdoor air had been circulated through the entire house during the night preceding the test. At 7:00 a.m. the windows were closed, and recirculation was begun with all of the air distributed to second story rooms. The condensing unit was started when the temperature on the second story reached 77.5 F.

A rapid reduction in relative humidity occurred on the second story when the condensing unit was started at 12:09 p.m. This was accompanied by a small reduction in dry-bulb temperature and resulted in a marked decrease in effective temperature. The relative humidity and dry-bulb temperature on the first story were affected in a manner similar to those on the second story, but to a somewhat lesser degree. The result was that a satisfactory effective temperature was obtained on both stories with cool air introduced only on the second story. The return air grilles were located on the first story in the two lower stair risers, and this location undoubtedly accentuated the natural tendency for the cool air to gravitate to the lower story. In general, the

[5] Loc. Cit. See Note 1.

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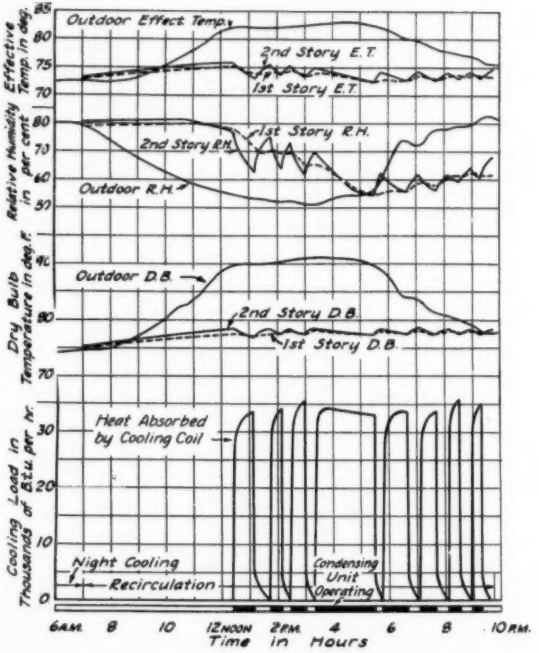

FIG. 6. GRAPHIC LOG OF AIR TEMPERATURES, RELATIVE HUMIDITIES AND COOLING LOAD. TEST NO 46. *Series* 4-38. AUGUST 23, 1938. CAPACITY 30,000 BTU PER HOUR. RECIRCULATION ON SECOND STORY ONLY

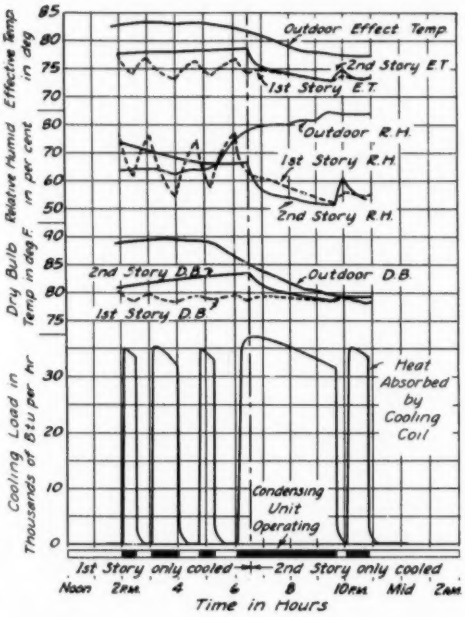

FIG. 7. GRAPHIC LOG OF AIR TEMPERATURES, RELATIVE HUMIDITIES AND COOLING LOAD. TEST NO. 40. *Series* 3a-38. AUGUST 9, 1938. CAPACITY 30,000 BTU PER HOUR. ALL AIR TO SECOND STORY AFTER 6:30 P.M.

results indicate that if partial cooling is to be used, it is advisable to locate the cool air outlets on the second story rather than the first, particularly if the return air grille is located on the first story. It is possible that in some cases first story rooms remote from the stairway may not be greatly affected. This did not seem to be true in the Research Residence, in which case the whole lower story was affected by the cooling on the second story. It was observed, however, that without positive circulation of air on the first story, any additions to the usual number of occupants, or any additional heat sources, soon produced unsatisfactory conditions in the rooms more remote from the return grille and the stairway.

Cooling on Second Story with Large Capacity Unit (*Series* 4-38)

The results obtained when the condensing unit was operated at a capacity of 30,000 Btu per hour and cooled air was delivered only to the rooms on the second story are shown in the third column of Table 3 and in Fig. 6. A schedule of operation similar to *Series* 2-38 was used, with air distributed only to second story rooms after 7:00 a.m. following the cooling with outdoor air during the night. However, since the capacity of 30,000 Btu per hour was ample to maintain a constant temperature in the house, before the condensing unit was started the temperature on the second story was allowed to rise to 78.5 F instead of the 77.5 F which was used with the smaller capacity in *Series* 2-38. On account of the high humidity conditions prevailing a starting temperature of 78.5 F on the second story was used instead of approximately 80 F which was used in 1934. With this starting temperature the house was maintained at an average temperature of approximately 78 F.

It may be noted from Fig. 6 that even the comparatively low starting temperature of 78.5 F was accompanied by a very high humidity which resulted in an effective temperature higher than 75 deg at the start of the test. Also the intermittent operation of the cooling plant produced large fluctuations in relative humidity and the effective temperature rose above 75 deg during the off-periods at the beginning of the test. For most of the test period, however, the conditions were entirely satisfactory. Since effective temperatures above 75 deg occurred during the off-periods, it follows that conditions in the house were more comfortable on hot days, when the plant operated with long on-periods, than they were on mild days, when the plant operated with short on-periods and more frequent off-periods.

During the entire test the temperature on the first story approximated the temperature on the second story quite closely. Humidity conditions were even better since, as shown in Fig. 6, the relative humidity on the first story was approximately equal to the average of the extremes occurring on the second story, and so resulted in more uniformly satisfactory effective temperatures. The effectiveness of cooling on the first story was subject to the same limitations as those discussed in connection with *Series* 2-38.

Rate of Reduction of Temperature on Second Story (*Series* 3-38)

During the summer of 1937 [8] a series of tests was made in which the first story was cooled during the day and the second story at night. The condensing

[8] Loc. Cit. See Note 1.

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unit operating at a capacity of 18,000 Btu per hour was used. When the cooled air was first distributed through the registers on the second story, the reduction in temperature and relative humidity was quite rapid. However, the reduction in temperature occurred very slowly after the first hour due to the effect of the heat absorbed in the structure, and amounted to 0.7 F per hour. The tests indicated that a comparatively large plant would be required to effect a rapid decrease in indoor air temperature.

In order to obtain comparative data on the reduction of temperatures and relative humidities, a similar series of tests was made in 1938 with the condensing unit operated at a capacity of 30,000 Btu per hour. The operating characteristics for a typical day are shown in Fig. 7. From 2:00 p.m. to 6:30 p.m. cool air was delivered only to the rooms on the first story and the temperature was maintained at approximately 79 F. During this period the temperature on the second story rose to 83.3 F. From 6:30 p.m. to 9:40 p.m. cool air was distributed only to the rooms on the second story and the air temperature was reduced 4.8 F. During this same period the relative humidity on the second story was reduced from 66 per cent to 51 per cent and the effective temperature was reduced from 78.6 deg to 73.0 deg.

The curves in Fig. 7 showing the rate of cooling on the second story have been reproduced (curves No. 1) in Fig. 8, together with similar curves from two other tests. The curves labeled No. 2 show the results obtained when the temperature on the second story was allowed to rise to 87 F before cooling was started. The broken-line curves designated as No. 3 have been reproduced from the 1937 series of tests with the condensing unit operating at a capacity of 18,000 Btu per hour.

It may be observed from curves Nos. 1 and 2 that the rate of cooling on the second story as measured by the decrease in dry-bulb temperature, relative humidity, and effective temperature was practically the same for starting temperatures of 83.3 F and 87.0 F. However, this increase in the temperature at which cooling was started lengthened the time required to lower the temperature on the second story to 80 F, resulting in a total time of five and one-quarter hours instead of one and one-quarter hours. Assuming that it is desirable to cool the second story to 80 F in not more than three hours, the maximum allowable temperature at the beginning of the period should not exceed 85 F with a condensing unit operating at a capacity of 30,000 Btu per hour. For initial air temperatures exceeding 85 F a larger capacity plant would be required.

When the condensing unit was operated at a capacity of 18,000 Btu per hour (curves No. 3), the temperature on the second story was reduced 4.0 F in the first 3 hours, indicating that this capacity was sufficient when the starting temperature did not exceed 84 F. The relative humidities during this period were higher than those maintained when the large capacity plant was operating, which in turn was reflected in higher effective temperatures. In general, the curves in Fig. 8 indicate that an increase in the capacity of the condensing unit from 18,000 to 30,000 Btu per hour did not materially increase the rate of cooling on the second story during the first hour of

operation, but did decrease the total time required to reach a predetermined temperature.

RESULTS OBTAINED WITH HEAT-FLOW METERS

The data obtained from the two heat-flow meter installations for a total of 20 days were plotted on a continuous chart which was used as a basis for the analysis of results. A typical section from the chart for the insulated section of the west wall, which was exposed to the sun in the afternoon, is shown in the lower portion of Fig. 9. From the 1937 data a similar day was selected

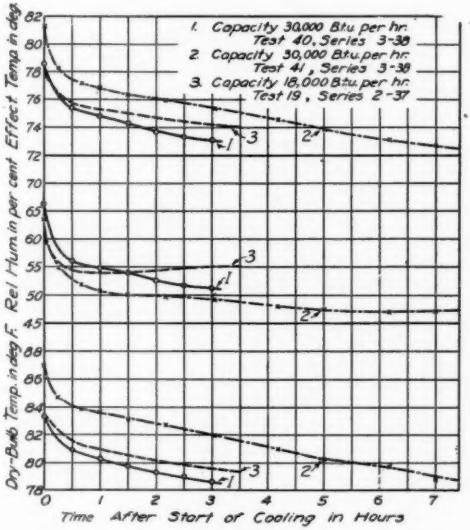
1. Capacity 30,000 Btu per hr. Test 40, Series 3-38
2. Capacity 30,000 Btu per hr. Test 41, Series 3-38
3. Capacity 18,000 Btu per hr. Test 19, Series 2-37
Dry-Bulb Temp. in deg. F. Rel. Hum. in per cent Effect. Temp. in deg.
Time After Start of Cooling in Hours

FIG. 8. RATE OF COOLING OF SECOND STORY

and the temperature curves for the uninsulated west wall were plotted, as shown in the upper part of Fig. 9. It may be noted that the temperatures were continually changing and that thermal equilibrium was never attained.

The two intermediate curves in Fig. 9 show the actual amounts of heat passing through the inside surface of the uninsulated and insulated walls, as indicated by the readings of the heat-flow meter. The meter was located directly below the thermocouples installed to obtain the temperature gradients through the wall. The maximum values of the heat actually transmitted into the room, as measured by the heat-flow meter, were 5.30 and 1.05 Btu per square foot per hour for the uninsulated and insulated walls respectively. The maximum rate of heat flow into the room for the uninsulated wall occured at 5:30 p.m., whereas that for the insulated wall occurred approximately four

hours later. The results obtained in 1937 indicated that the average time lag from maximum outdoor air temperature to maximum indoor air temperature was approximately two and one-half hours. The results from the 1938 tests indicate that this average time lag for the insulated wall was of the order of

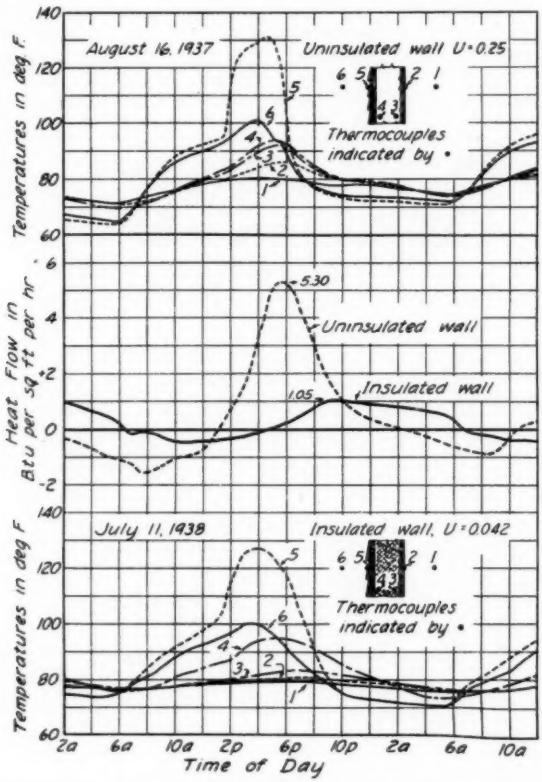

FIG. 9. GRAPHIC LOG OF TEMPERATURES AND HEAT FLOW FOR UNINSULATED AND INSULATED WALLS. RESEARCH RESIDENCE DATA FOR WEST WALL. SEASONS OF 1937 AND 1938

six and one-half hours. The total amount of heat per square foot transmitted into the room during a 24-hour period was obtained by integrating the area included between the curves of heat flow and the zero line. The values were 27.74 Btu per square foot for the uninsulated wall and 9.55 Btu per square foot for the insulated wall.

The curves shown in Fig. 10 represent average conditions based on data

obtained over a wide range of weather conditions. Hence, ratios based on these curves are more representative than similar ratios based on the individual days shown in Fig. 9. Both the data for the maximum rate of heat transmitted into the room and the total heat per square foot transmitted per day, as measured by the heat-flow meter, were plotted against the maximum value of the temperature of the outdoor surface of the wall. For a comparatively hot day in which the maximum surface temperature on the west wall was 130 F the ratio of the maximum rates of heat transmission for the uninsulated and

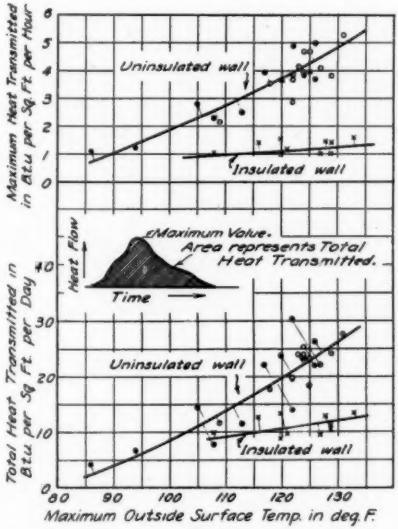
Maximum Heat Transmitted in Btu per Sq Ft. per Hour
Uninsulated wall
Insulated wall
Heat Flow
Maximum Value.
Area represents Total Heat Transmitted.
Time
Total Heat Transmitted in Btu per Sq Ft. per Day
Uninsulated wall
Insulated wall
Maximum Outside Surface Temp. in deg. F.

FIG. 10. MAXIMUM AND TOTAL HEAT TRANSMITTED PER DAY THROUGH UNINSULATED AND INSULATED WALLS. RESEARCH RESIDENCE DATA ON WEST WALL. SEASONS 1937 AND 1938

insulated walls was approximately four to one, as compared with a ratio of 6.0 to 1 based on the calculated coefficients of heat transmission. The curves for total heat transmitted per square foot through the west wall per day, as shown in the lower part of Fig. 10, indicate that the ratio of the total heats transmitted through the uninsulated and insulated walls for a hot day was approximately 2 to 1. In both cases for a milder day, in which the maximum surface temperature was less than 130 F, the ratios were smaller.

From a practical standpoint the curves indicate that the application of a substantial amount of insulation reduces the maximum heat flow through the walls by an amount only two-thirds of that indicated by the ratio of the calculated coefficients of heat transmission for the walls. However, the use of

insulation delays the time at which the heat input reaches a maximum value several hours beyond the time at which the design calculations of heat gain are generally made. Since the calculations of heat gain from a structure are based on maximum considerations, the fact that the maximum heat flow through

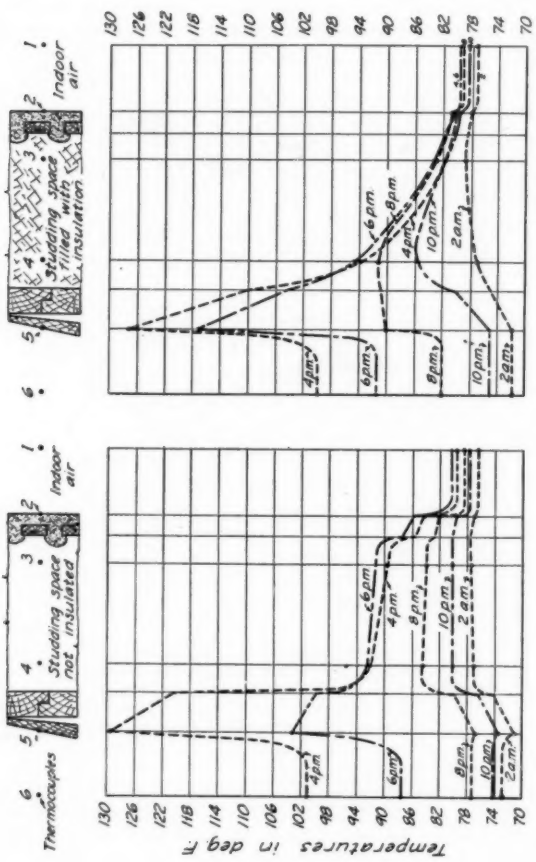

FIG. 11. TEMPERATURE GRADIENTS THROUGH UNINSULATED AND INSULATED WALL SECTIONS AT DIFFERENT TIMES OF THE DAY. DATA FROM WEST WALL. SEASONS OF 1937 AND 1938

an insulated wall occurs so much later than other maximum conditions indicates that some reconsideration of the methods employed in calculating heat gains in such structures may be advisable. In any event, the ratio of the maximum heat transmitted through the walls per square foot per hour has direct bearing on the relative sizes of plant required for structures with

uninsulated walls and those with insulated walls. On the other hand, the ratio of total heat transmitted through the walls per square foot per day has direct bearing on the relative cost of operation, once the proper size of plants has been selected. The results indicate that the apparent reduction in the maximum rate of heat flow through walls alone, as calculated from the coefficients of heat transmission, would be five-sixths, or 83.3 per cent, while the actual reduction would be three-fourths, or 75.0 per cent. That is, the actual reduction would be 90.0 per cent of the apparent reduction. Hence, if the heat gain through walls was 25 per cent of the total heat gain, the actual reduction in heat loss, and therefore the permissible reduction in plant size, to be effected by the use of insulation would be approximately 97 per cent of that calculated from the coefficients of heat transmission. In the case of the total heat transmitted daily through the walls the apparent reduction would again be five-sixths, or 83.3 per cent, but the actual reduction would be one-half, or 50 per cent, and the actual reduction would be 60 per cent of the apparent. In this case, if the heat gain through walls represented 25 per cent of the total, the actual reduction in daily heat gain, and hence the reduction in operating cost would be approximately 90 per cent of the apparent reduction as calculated from the coefficients of heat transmission. These results further indicate the necessity for additional investigation, probably under laboratory conditions, of the heat flow through walls as affected by cyclic variations in temperature on one or both sides.

The comfort conditions maintained in a given space are dependent not only on the temperature and humidity of the air, but also on the surface temperatures of the walls, to which the occupants are exposed. It may be observed from the curves labeled No. 2 in Fig. 9 that the inside surface temperatures of the exposed wall varied from hour to hour. For the purpose of comparison, temperatures taken from Fig. 9 have been replotted and shown as temperature gradients through the walls in Fig. 11. These temperature gradients indicate that the inside surface temperature of the uninsulated wall attained a maximum value of 86 F, while that for the insulated wall attained a maximum value of only 81 F, as compared with an indoor air temperature of approximately 78 F in both cases. Hence, with the same indoor air temperature, the insulated wall provided conditions which were the more suitable for comfort.

The temperature gradients for both the uninsulated and insulated walls indicate that at certain times of the day the temperatures in the studding space were higher than those of both the indoor and outdoor surfaces of the wall. That is, under these conditions the heat flow in the wall was occurring simultaneously to the outdoors and to the indoors. The calculated values of heat transmission are of little significance under such conditions; the significant values being given by the heat-flow meter, which on account of its location on the inside surface of the wall measures the actual flow of heat into the room. It may also be observed from a study of the readings of thermocouples Nos. 3 and 4, that, following a day on which the maximum outdoor temperature reached 100 F, the temperatures in the studding space were definitely reduced to a temperature approximating that of the indoor air. Hence, apparently there was no material storage of heat in the insulation to be carried over to the following day and to become cumulative in the case of a succession of hot days.

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SUMMARY AND CONCLUSIONS

The following summary and conclusions may be considered as applying to the Research Residence and the conditions under which the tests were conducted.

(1) In order to maintain the same effective temperature, the unit having the smaller capacity was operated to maintain an indoor dry-bulb temperature somewhat lower than that maintained with a unit having a larger capacity. Furthermore, it was advisable to allow for a rise in the indoor temperature on hot days when the cooling plant did not have sufficient capacity to maintain a constant temperature.

(2) The cooling capacity of 18,000 Btu per hour was found to be sufficient for both stories for days on which the maximum outdoor temperature rose to 95 F and the minimum outdoor temperature during the night preceding the test did not exceed 71 F. The daily median temperature (average of maximum and minimum outdoor temperature) of 83 F may be considered as the upper limit of the range of satisfactory application of the small-sized condensing unit.

(3) Subject to limitations over a small portion of the cooling season, the use of a condensing unit of small capacity was shown to be feasible for residences, especially in those cases in which the initial cost of installation is a more important factor than the maintenance of ideal indoor conditions.

(4) In the case of the Research Residence, satisfactory cooling on both stories was accomplished by the introduction of cooled air on the second story only. The application of this principle in general may be limited by the arrangement of the first story, particularly with respect to the location of the return air grilles and the location of rooms remote from the stairway.

(5) By introducing air on the second story only and increasing the capacity of the condensing unit from 18,000 to 30,000 Btu per hour the time required for cooling the second story from 83.5 F to 80 F was reduced from approximately 2¼ hours to 1¼ hours.

(6) The studies made with heat-flow meters indicated that considerable discrepancy existed between the measured values and the calculated values of the heat flow through the wall. Furthermore, thermal equilibrium was never attained in the wall section under actual conditions.

(7) In the case of the uninsulated wall the average time lag between the attainment of maximum temperature outdoors and maximum temperature indoors was approximately 2½ hours. In the case of the insulated wall this time lag was approximately 6½ hours.

(8) The application of insulation to a frame wall materially reduced the maximum hourly heat flow and the total heat flow through the wall per day. However, the reduction in both the maximum and total heat flow was less than that indicated by the ratio of the calculated coefficients of heat transmission of the uninsulated and insulated walls.

(9) The application of insulation reduced the temperature of the inside wall surface thus resulting in conditions more conducive to comfort for a given air temperature than those produced by the uninsulated wall.

(10) The insulated wall cooled to a temperature approximating that of the indoor air by 2:00 a.m., thus indicating that there was no material storage of heat in the insulation to be carried over to the following day and to become cumulative in the case of a succession of hot days.

ACKNOWLEDGMENTS

The results presented in this paper were obtained in connection with the summer cooling investigation (1938) in the Research Residence [7] at the Uni-

[7] The Research Residence in Urbana, Ill., was built, furnished and completely equipped specifically for research work in warm air heating by the *National Warm Air Heating and Air Conditioning Association* in December, 1924.

versity of Illinois, conducted by the Engineering Experiment Station of which M. L. Enger, dean of the College of Engineering, is the director, and in the Department of Mechanical Engineering of which O. A. Leutwiler, professor of Mechanical Engineering Design, is the head. These results will ultimately comprise part of a bulletin of the Engineering Experiment Station. Acknowledgment is also due to R. B. Engdahl, research graduate assistant, for services rendered in connection with the investigation, and to the various manufacturers who cooperated by furnishing instruments and equipment.

DISCUSSION

L. T. AVERY (WRITTEN): The authors are to be complimented on an unusually clear presentation of their material, with logical conclusions presented from the test data. They bring out the salient points of their investigation and the material will be of direct benefit to the industry, particularly in the effort to install simple and cheap cooling systems for the medium sized residence.

All of us in the air cooling work have noticed the apparent unsatisfactory result that comes from intermittent cooling. The authors point out that the larger machine operating intermittently did not produce a pleasing effect. This is equally true in commercial work and for that reason most of us prefer to use some means of unloading the machine to get two or more steps of operation. It is practically impossible to set a thermostat close enough to hold to the top of the effective temperature range without permitting overheating before the machine starts up and if the thermostat is set lower there is considerable over-cooling before the machine shuts down. In our office we find that the two-speed motor on a 3 hp machine operating from a two-point effective temperature thermostat gives us fairly uniform conditions, much more pleasant than can be secured with a single speed machine which turns all off or all on.

We also wish to confirm the observations as to the effect of hot walls. If we have Venetian blinds closed tight to reflect the sun rays, these give the effect of panel heaters, although the effective temperature may be well within the comfort limits. There is a very decided radiant heating effect which makes the occupant uncomfortably warm. We have noticed this same effect in restaurants where patrons sit near steam tables or coffee urns. The paper emphasizes the importance of keeping radiant heat from the conditioned space and in a residence the sun heat will be the principal difficulty.

It seems to me that we are overemphasizing the importance of insufficient cooling. Undoubtedly the manufacturers and sales offices will welcome this paper as offering encouragement to sell 1½ hp refrigerating units to cool a residence. I suspect that as long as we take the attitude that a little cooling is better than none, we will hold back the acceptance of air conditioning by the public.

During the past 20 years I have seen the customer improve his own standards of comparison and he is now just as critical of cooling that is too cold as he used to be of cooling that was inadequate. It is true we have educated the public to accept 77 F to 79 F with a satisfactory humidity as against his old misconception of 72 F, but we cannot depend on the open window at night to help out the small machine in the ordinary home. A. H. Tashjian of Cleveland has designed an automatic window opener and a device of this kind could be hooked into your control circuit so that when the outside effective temperature reached the proper point all the windows would be opened and again when the outside temperature rose the windows would be closed. What is going to keep out the summer rain storm, the dust and

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the pollen? As along as this industry tries to compromise with the open windows, just so long will the open-window advocate hold us in a secondary position.

It is true a great deal of winter heating is wasted by open windows and the year-round conditioning system will save in the winter time due to its ability to economize on what is otherwise wasted heat. It is not consistent, therefore, to revert to the open window for summer cooling, attractive as it may seem. Outside air for cooling, yes. Certainly we have often cut the outside air allowance too thin for good ventilation results and just that way we have cut down the ability of outside air to help us with the cooling, but outside air must be brought through the filters to remove dirt and pollen and the windows must be kept closed to prevent worry and damage from the sudden rain storm. The air conditioning equipment can then be depended upon to do the job it was bought for.

We can, therefore, be careful in accepting the data presented by this paper as indicating a 1½ ton machine sufficient for a residence of this size and type. The man who buys or builds a $15,000 home probably spends $300 a year for a good automobile. The operating cost for summer conditioning is insignificant compared with the other expenses to live under present conditions. No one is going to thank us for selling a $400 cooling system when a $700 system is needed. Summer air conditioning will be accepted and paid for at a fair price when we have the courage of our convictions to insist that every sale include the equipment and the design to assure satisfactory results.

T. H. URDAHL (WRITTEN): The authors of this paper deserve especial credit for having further illuminated the properties of insulation as these apply to comfort and to the economy of modern structures of all types which are designed to meet the demands for maximum comfort and efficiency. These properties observed through the measurement of heat flow and lag through insulated and non-insulated walls serve to emphasize the importance of insulation as a factor in producing comfort in cooled buildings and in determining equipment capacities. Also, and perhaps of greater importance, in lowering operating costs.

When considered from the standpoint of heating, insulation has already been proven out of the luxury class into an economic necessity as a component of sound construction and now, in demonstrating its properties for reducing heat gain it further stabilizes its position as an essential. To provide the advantages of comfortable temperature conditions to the greatest majority, initial and operating costs far below the present level are required. The data presented concerning the effects of insulation and the use of small capacity equipment serve as a guide toward the accomplishment of these ends.

In applying the theorems developed through these tests I find that they parallel experience obtained in an office building in Washington, D. C., during the past seven years. This structure of reinforced concrete frame with 12 in. outside brick curtain wall construction furred and plastered on the inside, was equipped with a cooling system approximately 40 per cent under actual design requirements. The design of the cooling system was not adapted to the use of outside air for night cooling, nor would this have been feasible at all times since high temperatures and humidities prevail by day and night for periods of long duration. Approximately 50 per cent of the total actual load was in internal heat such as lights, occupants, etc., and it was frequently necessary to supplement plant capacity with ice in order to stay within even higher effective temperature limits during peak hours. Due to the heat storage in outside walls and the lag in heat transmission of about 8 hours, continuous operation over 24 hours was necessary. Except in very cool weather a shut-down of the plant was only due to a breakdown of the machinery and it was useless to begin its operation until after a period of cool weather reduced the building temperature to the degree which the plant could maintain. Thus with capacity barely sufficient to satisfy internal and solar heat gains through shaded windows, all night operation was re-

quired to remove accumulated heat from the peak carry over after 6:00 p.m. and the transmission through walls.

In 1937 a plant of adequate capacity was installed resulting in the ability to maintain comfort conditions at all times with day time operation only and a total of 1200 hours operation per season, compared with 3100 hours per season for the smaller plant. On the basis of calculated heat gain reduction through insulation of outside walls and roof with certain assumed factors of performance which check closely with those obtained by the authors of this paper, the smaller plant would have served adequately with only a 12 per cent increase in the hours of operation over those now required with the larger plant.

The purposes of use and hours of occupancy of any structure and the consequent value of applying of insulation should receive the first consideration of the designer, especially since the values in the cooling equation are higher due to the higher unit cost of electrical energy and of cooling equipment.

Financial interests to whom modern structures represent security for investments will give added assurance to the soundness of these by insisting that the thermal properties of construction materials employed are in keeping with the accepted requirements of year-round comfort for human efficiency, thus avoiding waste of costly energy and disproportionate investment of capital in needlessly large and proportionately more rapidly depreciating machinery.

Since the characteristics of climatic conditions vary in different sections of the country in a manner so inconsistent as to make an interpolation of data of doubtful value, I recommend that work similar to that which has been done by the authors of this paper be carried on under their direction in localities representative of other conditions.

A. B. NEWTON (WRITTEN): The authors are to be congratulated in the completeness of detail in reporting their work as well as the number of conditions set up for test.

I am particularly interested in the graphical comparison between the results with a condensing unit capacity of 18,000 Btu per hour and the results at 30,000 Btu per hour. This comparison seems to me to illustrate very definitely one of the reasons why an obviously small unit sometimes produces a more acceptable comfort result than does a larger unit. If Fig. 5 is compared with Fig. 6 it will be seen that while cooling was in progress the smaller unit in Fig. 5 operated continuously while the larger unit of Fig. 6 cycled *on* and *off* several times. As a result of this cycling action some variation in temperature occurred. The dry-bulb variation was accompanied by a large variation of relative humidity, and since these two factors rise and fall together they produce a decided fluctuation in effective temperature.

It might be expected that under milder conditions even the low capacity units would cycle *on* and *off*, though less frequently than the high capacity unit. I believe any comments the authors may be able to make regarding the effect of these fluctuations on comfort, would prove interesting. In my own experience it has been indicated that the cyclical variations in temperature and humidity, particularly the latter, are an important contributing factor in producing uncomfortable *comfort cooling*.

These cycles may be largely eliminated by gradually reducing the capacity of the refrigeration system as the house temperature falls, and thus maintaining the compressor in almost continuous operation as soon as cooling is needed. If a constant speed compressor is employed, this reduction is quite generally secured by throttling either the liquid or the suction line. It should be noted that throttling the liquid line results in lowered coil temperature at reduced load, so that generally the relative humidity may be held lower than when throttling the suction line and thus raising coil temperature.

Since these factors appear to have a very definite bearing on the degree of comfort

produced in summer conditioning, I believe a comparative series of tests at the Research Residence might be in order if the contemplated program will permit.

JOHN EVERETTS, JR. (WRITTEN): I feel that with the calculation load of 30,000 Btu per hour for this particular residence that in order to maintain the conditions under which this load was calculated it would require a cooling system of 30,000 Btu capacity, and to use a system any smaller than that is, in my opinion, a waste of time and money as it will not do the cooling required.

If conditions other than those used in the basic calculations are satisfactory then the calculations should be designed on those conditions which will change the heat load and consequently allow for a smaller size system.

I believe that this committee could do constructive work by making a study of heat lag and effect of rain and wind upon the over-all operation of a given system.

C. F. CUSHING: This and other papers by Professor Kratz and his associates, reporting the summer cooling test results in the Research Residence, are very valuable to those interested in the promotion of residential summer conditioning, as they greatly facilitate the work of the application engineer in offering the prospective user a low-cost installation that will provide satisfactory comfort conditions.

It is quite probable that more desirable conditions might be attained if the equipment incorporated some means for direct dehumidification, together with a larger capacity for moisture removal. The wide fluctuations in relative humidity which occurred with the cycling of the compressor would be eliminated, and lower dew-points, and consequently lower effective temperatures, could be maintained.

The test results indicate that if the authors had attempted to maintain the indoor dew-point at approximately 60 F and hence relative humidities between 50 per cent and 55 per cent, equipment having a 50 per cent latent removal capacity would have been required. On this basis a dehumidifier of approximately two tons capacity, followed by a resaturation spray cooler designed for one ton of adiabatic transfer, would probably have produced very good results. Such an arrangement would provide independent control of humidity and temperature, as well as maintain lower dew-points, regardless of how the proportion of load might fluctuate.

Consequently, if the authors would supplement their valuable contributions with similar experiments using direct dehumidifying equipment and resaturation cooling, I'm sure the results would be most interesting and informative.

AUTHORS' CLOSURE: Mr. Avery mentions that this paper might possibly be used as a basis for advocating the use of undersized cooling plants. It has not been the intention of the authors to advocate anything. It is the function of the Engineering Experiment Station to investigate problems and report the facts impartially. Any advocacy of special methods and apparatus, or of pet theories is contradictory to such an expressed purpose. In this case it was recognized that undersized plants are sometimes installed either by accident or by design. The purpose of these tests, therefore, was to determine the limitations of the undersized plant on the basis that if such a plant were to be installed, either intentionally or otherwise, the results presented would serve as a source of information concerning the limitations to be expected.

Mr. Newton asked for comments on the effect of the cycling operations on comfort. The authors discussed this matter in a paper[8] previously published, in which the desirability of some modulating method was mentioned.

[8] Loc. Cit. See Note 1.

No. 1138

COOLING TOWER PERFORMANCE STUDIES

By L. M. K. BOELTER,* BERKELEY, CALIF.

This report (Part 1) is the result of cooperative research sponsored by the Committee on Research of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and conducted at the University of California, Berkeley, Calif.

INTRODUCTION

THIS report presents data indicative of the direction in which the Evaporative Cooling Project (cooling towers) is proceeding at the University of California as outlined by the ASHVE Technical Advisory Committee on Cooling Towers, Evaporative Condensers and Spray Ponds: B. M. Woods, Chairman; J. C. Albright, S. C. Coey, E. H. Hyde, E. H. Kendall, S. R. Lewis, J. Lichtenstein, J. F. Park and E. H. Taze. Further instrumentation is required and is in progress. The tentative results on drop dynamics will be augmented by the application of these results to the mass and heat transfer phenomena. These results will be available by December 1939 and their application to the spray tower will be undertaken. Computations on the effect of drop size on vapor pressure, small drop thermal stability, liquid interface cooling due to evaporation and vapor pressure reduction due to solvents (impurities) are under consideration. Tests on the atmospheric towers were in progress during the summer of 1939.

Description of Cooling Towers

The cooling towers available for test are shown in Figs. 1 to 6 and will be described briefly:

1. Cooling Tower No. 1: Atmospheric type model of multiple-section cooling tower with general characteristics as follows: Height 11 ft; width 4 ft; length 9 ft over-all (composed of two 4-ft sections).

Nine decks plus the distribution deck are spaced 1 ft apart. Distribution is accomplished by means of a central longitudinally-placed trough with small v-notch weirs along the upper edges.

The slats are ⅜ x 3⅝ in. x 3 ft 8 in., laid with their length parallel to the greatest dimension of the tower. The slat mats are arranged with six slats on the upper side and seven slats on the lower side across the width of the tower.

* Professor of Mechanical Engineering, University of California.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Mackinac Island, Mich., July, 1939.

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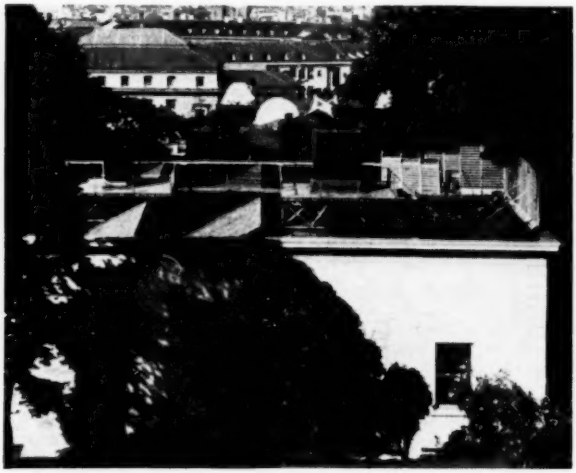

FIG. 1. VIEW OF ROOF OF HESSE HALL REVEALING LOUVERS AND SPRAY POND

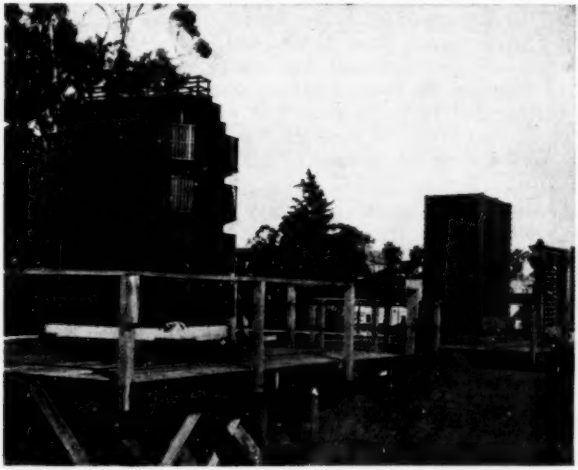

FIG. 2. VIEW OF TOWERS NO. 1 (*right*) AND NO. 2 (*left*) ON ROOF OF HESSE HALL

The louvers are at an angle of 60 deg with the horizontal. There are two louvers for each deck interspace.

2. Cooling Tower No. 2: Atmospheric type one-quarter full size scale model: Height 9 ft; length 6 ft; width 3 ft.

There are four aerator louver sections and a distributing section. Eleven decks plus the distribution deck are spaced at 8 in. Slats are ¾ x 3/32 x 17 in. redwood.

Each deck is composed of eight 17 in. square sections. Each deck section contains 12 slats on the upper side and 13 slats on the lower side. Alternate decks are placed with the slats in opposite directions.

Distribution is accomplished by means of spray nozzles which discharge upward against special splash plates.

Approximate rating: 2 gpm ft² deck area; wind 3 mph; air at 70 F WB temperature; water cooling range = 10 F; approach to wet-bulb is then approximately 8 F.

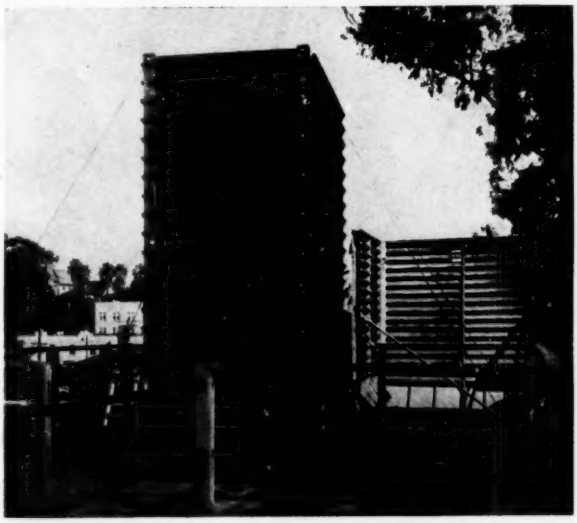

FIG. 3. ATMOSPHERIC TOWER NO. 1

3. Cooling Tower No. 3: Forced Draft Spray Type. This tower is of sheet metal construction and pertinent data are as follows: Height 7 ft; cross-section 3 ft x 4 ft over-all (composed of two 2 ft x 3 ft chambers).

The design is such that the air first passes in parallel flow with the water downward through a 7 ft x 3 ft x 2 ft spray chamber and then passes upward in counter-flow with the water through a similar chamber.

There are six spray nozzles in each section.

The forced-draft fan is driven by a ⅓ hp motor and is rated at 2000 cfm (actual capacity up to 3000 cfm).

Rating: Capacity = 3 tons; cooling range = 7 F.

4. Cooling Tower No. 4: Induced Draft Packed Type. This is an induced-draft counter-flow tower packed with special ovate wooden slats placed vertically. Pertinent data are as follows:

Height 12 ft, including a 3-ft drop-out chamber between the upper deck and the

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fan; cross section 3 ft x 3 ft. The casing is made of redwood shiplap, $^{25}/_{32}$ in. x 5 in. face.

There are 22 decks composed of ovate slats with 12 slats per deck and 13 slats per deck staggered in alternate layers. The ovate slats are 2¾ in. high by ⅞ in. maximum width.

Distribution is accomplished by means of a 3-in. I.D. wooden pipe across the center of the tower. This pipe runs transversely with respect to the direction of the slats, and holes are bored on the horizontal diameter directly above each distribution slat.

Performance data are available in the form of test results which include the following conditions: Air rate = 3 lb per second of dry air; water rate = 3.5 lb

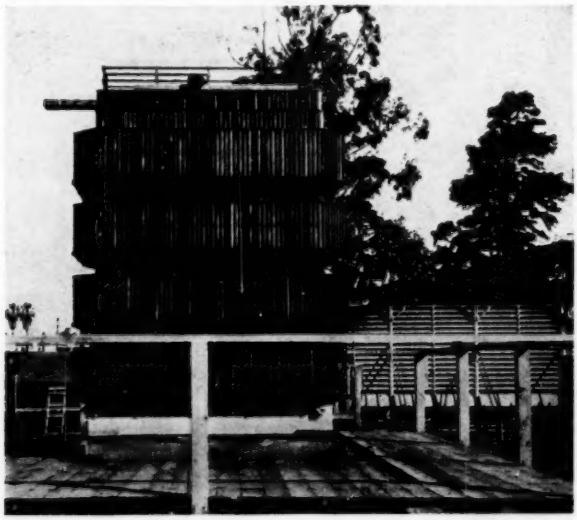

FIG. 4. SCALE MODEL ATMOSPHERIC TOWER NO. 2

per second, or 3.06 gpm (square foot of cross section); dry-bulb temperature of entering air = 70 F; wet-bulb temperature of entering air = 60 F; water temperature in = 85 F; water temperature out = 75 F.

The roof of Hesse Hall is partially protected by louvers and an extension of the louver system is contemplated. The atmospheric towers have been placed on the roof of Hesse Hall (as shown in Figs. 1 and 2) while the forced-draft types were placed inside of the same building. The piping arrangement for all of the towers is interconnected and is diagrammed in Fig. 7.

In addition to the tower facilities a spray pond equipped with high and low pressure nozzles is available on the roof of Hesse Hall. Data have been taken on this spray pond but they have not yet been evaluated.

DESCRIPTION OF INSTRUMENTS DESIGNED AND BUILT

1. Hair Hygrometer. This instrument offers some advantages over other types for the determination of the space distribution of humidity. The design proposed by Büttner [1] is shown in Fig. 8 together with a typical calibration curve. A time lag inherent in the instrument precludes its utilization for atmospheric conditions in which the humidity varies rapidly with time. The

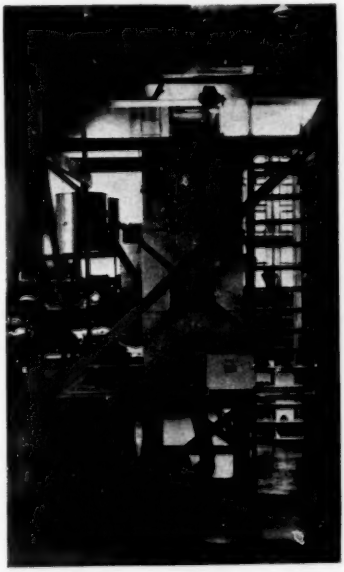

FIG. 5. FORCED CONVECTION SPRAY TOWER NO. 3

variation of length of the human hair is amplified by an 0.011 in. diameter cantilever, the end deflection of which is measured by means of a microscope.

2. Thermocouple Psychrometer. Precision space psychrometry [2] demands a wick diameter which will provide adequate cooling to the wet-bulb temperature at the local air velocity. Often the free convection generated by the wet junction depression is the only circulation available. Powell [3] and Cammerer [4]

[1] Büttner, K. Die Wärmeübertragung durch leitung und Konvection, Verdunstung, und Strahlung in Bioklimatologie und Meteorologie. *Abhandlungen, Prussian Meteorological Institute*, Vol. X, No. 5, p. 1-37 (1934).
[2] Walker, Lewis, McAdams and Gilliland, *Principles of Chemical Engineering*, p. 585, (1937).
[3] Powell, R. W., *Proc. Physical Society of London*, 48, 406-14, (1936).
[4] Cammerer, J. S., *Die Messtechnic*, Vol. XIII, No. 2, p. 21, (1937).

indicated the design limits of the thermocouple psychrometer. A unit designed for free convection circulation will possess a wick diameter less than 1 mm. In the type illustrated in Fig. 9, No. 36 constantan wire is plated with copper throughout part of its length thus providing two junctions. The length of the thermocouple wire must be sufficiently great to reduce conduction to the wet junction to a minimum. Either the wet junction and dry junction voltages or the wet junction depression and the dry junction voltages may be measured.

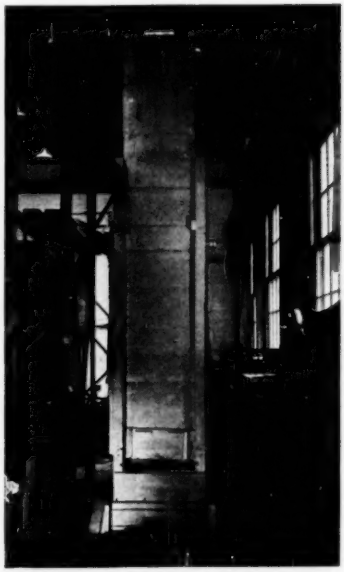

FIG. 6. PACKED TOWER NO. 4

The wick is either of dental floss or cotton properly boiled and is fed from a glass reservoir.

3. Absolute Hygrometer. In order that the water vapor content could be determined directly an absorption hygrometer (Fig. 10) was designed after Blackie.[5] The sample of moist air is drawn into the thermally insulated volumetric chamber *A* by reducing the mercury level to *J* by means of the leveling bottle *M*. Cock *C* is then opened to the sulfuric acid pipette *F* (the initial acid level being at *E*). Raising *M* then fills Chamber *A* with mercury and the air sample is displaced into the drying chamber *F*. Upon complete absorption of the moisture the acid level is restored to *E*, the mercury level to *J*, and the pressure on the manometer is read by means of the scale *K*

[5] Blackie, A., *Journal of Scientific Instruments*, 13, 6, (1936).

Upon correction for the air temperature change (indicated by thermometer *B*) the difference between the initial and final pressure readings will be the partial pressure of the water vapor.

4. Radiometer. In order to measure the rate of energy radiated by the tower under operating conditions as well as to determine the rate of irradiation of the tower, a portable radiometer illustrated in Fig. 11 will be utilized. Briefly, the unit consists of a multi-junction receiving unit placed at the apex of a conical aperture. The aperture is defined by baffles attached to a high heat capacity housing; blackened on the inside, chromium plated on the outside. The receiving element consists of 140 silver-constantan junctions, wound 104

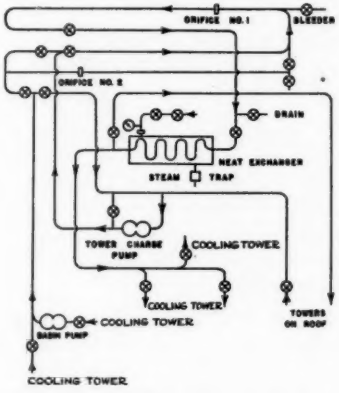

FIG. 7. PIPING DIAGRAM FOR COOLING TOWER LIQUID (WATER) DISTRIBUTION

per inch. The constantan is wound on a frame (thermal and electrical insulator) and one half of the winding is silver plated. The receiving junctions are covered with a *tin-foil* blackened strip. The inverse response (sensitivity) of this radiometer is 7.5 $\frac{\text{Btu}}{\text{foot}^2 \text{ hour mv}}$. The unit is frequently calibrated either by means of a hohlraum, a blackened surface at a known temperature or a thermal standard.

A pyroheliometer is available to measure solar and sky radiation.

5. Anemometer. A pressure plate anemometer [6] illustrated in Fig. 12 has been designed and partially constructed. The motion of a diaphragm, excited by the air stream, and resisted by a spring is transmitted to a portion of the core of a coil (transmitter) thus changing the reluctance of the magnetic

[6] Sherlock, R. H., and Stout, M. B., An Anemometer for a Study of Wind Gusts. (University of Michigan, *Engineering Research Bulletin* No. 20, May, 1931.)

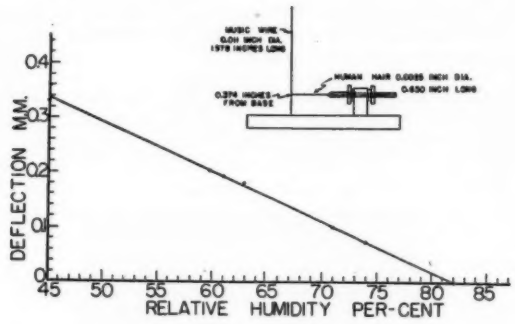

FIG. 8. HAIR HYGROMETER AFTER BÜTTNER AND THE CALIBRATION THEREOF

circuit. A fixed reactor forms another leg of a bridge. Proper instrumentation will yield either the rms air velocity or its instantaneous value. The natural frequency of the moving parts is sufficiently high to measure the velocity fluctuations which will be effective in the cooling of a water stream.

Hot wire anemometers have also been designed for this work. A cup anemometer of special design yields the usual velocity data. The details of these units will be reported later.

6. Photography of Drops. The behavior of the drops within the tower will be studied stroboscopically. Special apparatus has been designed to photograph those particles which constitute drift. The apparatus shown in Fig. 13 is due to Gullberg. The particles are collected on a nitrate dope surface placed at *O*, and may be observed and photographed simultaneously. Photography may be

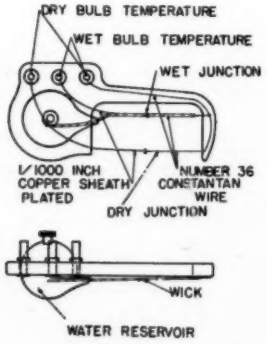

FIG. 9. THERMOCOUPLE PSYCHROMETER

accomplished under conditions of illumination which will not cause drop evaporation.

Discussion of Operating Principles

The internal operation of a cooling tower is based entirely upon surface phenomena. The progressive disintegration of the liquid stream, which has been introduced into the tower, into many jets and small drops is governed by the principles of surface energy (surface tension), surface dynamics (capillary vibrations and considerations of stability), and the effect of the

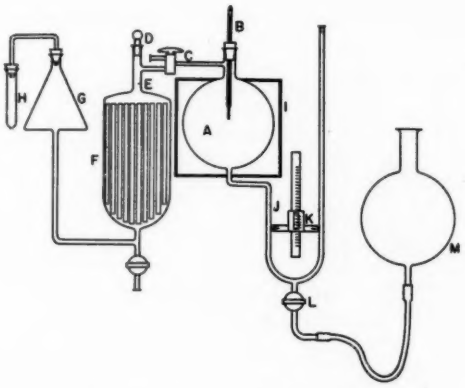

FIG. 10. ABSOLUTE HYGROMETER

A, Sampling chamber; *B*, Thermometer; *C*, Three-way cock; *D*, Stopper; *E*, Datum mark; *F*, Drying chamber; *G*, Reservoir; *H*, Calcium chloride tube; *I*, Aluminum foil screen; *J*, Datum mark; *K*, Sliding indicator; *L*, Cock; *M*, Mercury reservoir.

frictional drag of the air currents on the surface of the falling water. The liquid wets and spreads on the internal surfaces of the tower as a consequence of surface adsorption and the action of the interfacial surface tension; and once the surfaces are wet, a diffusion of some of the liquid across the surface and into the material (if porous) from which the tower has been made takes place. The shattering and subsequent distribution of the jets and drops impinging upon the internal surfaces of the tower depend upon the microscopic characteristics of these surfaces and their arrangement.

The first phase of the evaporative cooling project will include a study of cooling tower behavior. Over-all performance data will be obtained for the several distinctive types of towers described earlier in this report. The behavior of these towers under different operating conditions will serve to reveal the different phases of the general problem which deserve particular experimental and analytical attention. The component unit processes which constitute the operation of a cooling tower will be outlined later. The end of the research

project will be to present data and correlations which will make possible the rational design of a tower in terms of the independent variables. Some of these data are now available—others need to be determined. The data will be obtained and will apply to ideal systems, the composite of which will constitute the complete ideal tower. The actual tower performance should differ but little from the ideal tower performance if the basic design data are comprehensive and accurate.

When a cool unsaturated gas passes over the surface of a hot liquid, a simultaneous exchange of heat and material takes place by molecular mechanisms across the interphase boundary. Formulation of the resultant effect of these superimposed processes in terms of physical laws will lead to the only satisfactory methods of interpreting test results. The cooling tower may be considered as an extraction column in which one component is missing. The

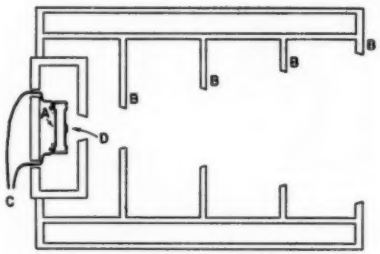

FIG. 11. PORTABLE RADIOMETER

A, Silver-constantan thermopile; *B*, Baffles; *C*, Lead-in-wires; *D*, Receiver.

results of the test made on the towers will be presented in a form similar to those used to describe the behavior of absorption and extraction columns.

The cooling process as it proceeds in the tower consists almost entirely of surface transfer phenomena. The liquid stream is progressively broken into jets, drops and sheets. In packed towers the liquid wets the internal surfaces, the resultant shattering of the drops, jets and sheets being dependent upon the surface characteristics of the packing. Evaporative cooling is directly dependent upon the phenomenon of surface diffusion, which involves considerations of surface configuration, rates of interfacial mass transfer, and surface concentrations and concentration gradients. Cooling by direct heat transfer, upon introduction of concepts of the rate of interfacial transfer of thermal energy by conduction and convection and surface radiation, is directly dependent upon the surface configuration, surface temperatures and temperature gradients at the surfaces, and the radiant emissivities of the surfaces. In addition, there are the perhaps inappreciable but nevertheless not-to-be-overlooked effects of the heats of surface adsorption, wetting and dilution, cooling effects which accompany the stretching of surface films, static electrical charges on the liquid and tower surfaces, changes in surface areas due to thermal expansion, chemical reactions which may take place on any of the boundary

surfaces of the liquid-solid-gas system which exists within the cooling tower, and impurities which collect on the tower and liquid surfaces—(such as dust, oil, slime or solutes).

Comparatively little detailed information is available concerning phenomena of this type. And it is largely due to the lack of quantitative (or qualitative in some cases) information concerning these microscopic details and their relation to the over-all performance of cooling towers under different operating conditions that many uncertainties have arisen in connection with a complete understanding of the design and operation of present-day semi-empirically constructed towers.

Passing to a consideration of the relation between the internal operation and the many independent external variables, it is seen that for a given cooling duty of an atmospheric tower the constantly changing weather conditions and

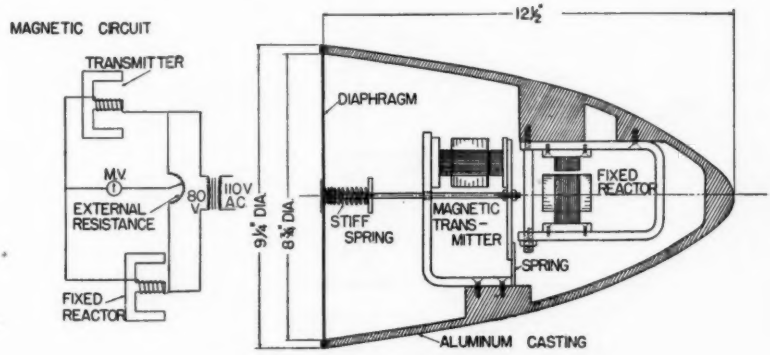

FIG. 12. PRESSURE PLATE ANEMOMETER

the rate at which cooling must be accomplished represent the independent variables to which operating conditions within a given tower must be adjusted.

In order to present an adequate formulation of the problem, it is necessary to assemble for consideration all of the variables which can affect the performance of a cooling tower in any way. In doing this, it would seem to be most significant to take a broad point of view and consider a *cooling tower* to be any structure which is operated for the purpose of cooling *any* liquid primarily by the method of evaporating a part of the liquid stream into a stream of gas as the two fluids flow through the structure.

Some of the variables noted here may be negligible in effect. In order to accomplish a fairly complete presentation, the list has not been restricted to the independent variables. Furthermore, all variables are to be considered in the light of different operating conditions in any given tower. The relative effects of the variables with different types of construction and with different fluids should be investigated together with the effects of the structural details themselves.

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A possible grouping of the variables follows:

A. THE GAS CIRCUIT

1. The gas temperature and the variation throughout the structure.
2. The moisture content of the gas and the variation throughout the structure.
3. The barometric (total) pressure and the pressure variations throughout the structure.
4. The physical properties of the gas which have a bearing on tower performance and which commonly vary with the temperature, pressure and moisture content:—density, specific heat, thermal conductivity, viscosity, coefficient of thermal ex-

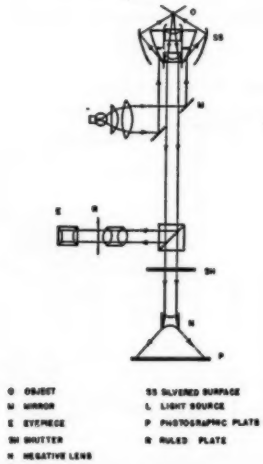


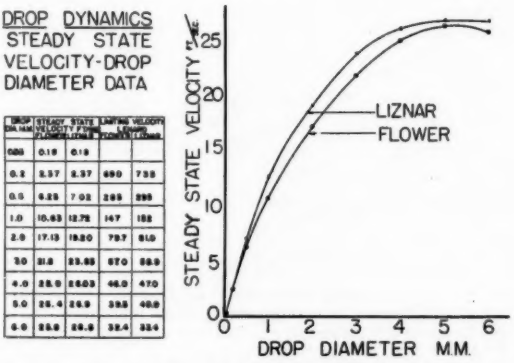
FIG. 13. DRIFT PHOTOGRAPHY-ARRANGEMENT OF OPTICAL EQUIPMENT

pansion, vapor diffusivity, apparent molecular weight and gas constant, number and type of gaseous ions present.

5. The rate of flow, over-all and in different sections of the tower.
   a. Velocity distribution and turbulence within the tower as affected by the type of construction, the rates of liquid and gas flow, and the weather conditions (initial conditions in the gas stream).
   b. Flow resistance characteristics of the tower passages and surfaces.
6. The weather variables. (The word *weather* as used here should be considered as a general term referring to the initial conditions in the gas stream. The relative importance of some of the variables in this category obviously depends largely upon the type of tower construction involved.)
   a. Temperature and its variation.
   b. Wind velocity and direction as periodic and transient functions of time.
   c. Solar (nocturnal) radiation.
   d. Humidity and its variation with related conditions.
   e. Presence of dust and other impurities.

B. THE LIQUID CIRCUIT

1. The liquid temperature and the manner of variation throughout the structure.
2. The physical properties of the liquid which have a bearing upon tower performance and which depend upon the liquid temperature: Density, specific heat, thermal conductivity, viscosity, vapor pressure, surface tension, latent heat of vaporization, coefficient of thermal expansion, surface emissivity—(Radiant energy exchange), molecular weight and gas constant of the vapor, vapor diffusivity.
3. The rate of flow and the size of the sheets, jets, and drop. (The cooling duty as related to the form and distribution of the liquid-gas surfaces of contact and the dynamical and thermodynamical conditions there existing.)
   a. Pressure and elevation head in the liquid supply system and details of the construction and installation of the supply and distribution system and its discharge characteristics.
   b. Liquid distribution within the tower as affected by the other variables. Degree



DROP DIA. MM.	STEADY STATE VELOCITY FT./SEC.		LIMITING VELOCITY FT./SEC.	
	FLOWER	LIZNAR	FLOWER	LIZNAR
0.05	0.19	0.19		
0.1	2.57	2.37	690	732
0.5	6.23	7.02	285	295
1.0	10.63	12.72	147	155
2.0	17.13	19.20	79.7	81.5
3.0	21.5	23.65	57.0	58.9
4.0	23.0	26.03	46.0	47.0
5.0	25.4	28.9	38.5	40.0
6.0	28.0	29.8	32.4	33.4

FIG. 14. DROP DYNAMICS. STEADY STATE VELOCITY AS A FUNCTION OF WATER DROP DIAMETER. DATA OF LIZNAR AND FLOWER

of turbulence and the amount of free surface. Relative value of the different constructional details.
   c. The frictional drag of the gas on the liquid surfaces. Effect of fluctuations in velocity.
   d. Of prime importance are all conditions having a bearing upon the stability and dynamics of the liquid surfaces.
4. Adsorption of the liquid films by the tower surfaces.
   a. Microscopic surface characteristics. Interfacial surface tension.
   b. Effect of slime and solid deposits on the tower surfaces.
   c. Effect of paint on the tower surfaces.
   d. Diffusion of the liquid into the tower material.
5. Effect of dissolved gases and other impurities in the liquid upon the cooling process. (Modification of dynamical and thermodynamical conditions.)
6. Cooling or heating effects in the liquid circuit leading to or from the tower.

C. CHARACTERISTICS OF THE STRUCTURE ITSELF AND THE MANNER IN WHICH IT IS ARRANGED AND OPERATED

1. Constructional details.
   a. Dimensions, relative proportions.

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b. Materials.
c. Arrangement and type of the internal surfaces.
d. Louver design. Flow paths for gas stream. Relation to the drift loss.
e. Liquid supply and distribution system.
f. Levelness of all surfaces.

2. The temperature of the structure itself at different points as influenced by external and internal conditions.

3. Loss of liquid by entrainment. (Drift loss.) Effect of constructional details and weather conditions.

4. Design and location of the basin at the bottom of the tower. Surface evaporation from this basin.

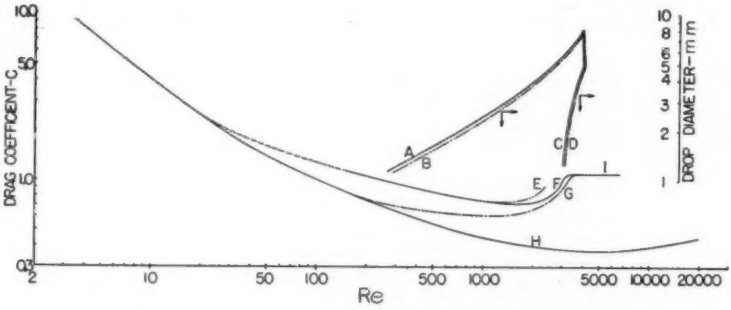

FIG. 15. DROP DYNAMICS

STEADY STATE REYNOLDS' NUMBER
PLOTTED AS A FUNCTION OF DROP DIAMETER

A Flower and Wieselsberger
B Liznar and Wieselsberger

LIMITING REYNOLDS' NUMBER FOR STABLE DROP
AS A FUNCTION OF DROP SIZE

C Lenard and Flower
D Lenard and Liznar

DRAG COEFFICIENT AS A FUNCTION OF REYNOLDS' NUMBER

E	Methyl salicylate drop	—Flower
F	Water drops	—Flower
G	Water drops	—Liznar
H	Spheres	—Wieselsberger
I	Flat plates	—Wieselsberger

5. Structural vibrations caused by the wind, the impact of the falling water, and extraneous effects,—*i.e.*, vibrations caused by nearby machinery or vibrations of the supporting structure if the tower is mounted on or in a building.

6. Noise generated by the air, water and structure vibration.

7. Location of the tower.
a. Heat transfer to and from the surroundings,—especially by direct and reflected radiation.
b. Orientation with respect to the prevailing wind velocity.
c. Interference effects of surroundings.
d. Drift loss as a nuisance.

8. Type of gas circulation provided for; *i.e.*, free or forced convection in any of the many special arrangements.

9. An important general item in this connection is a special study of the importance of the time and relative direction of contact of the gas and liquid and the inter-

facial surface areas and the resultant effects on tower performance as influenced by constructional details and arrangements and the operating conditions as determined by the many independent variables.

For example, the unit mass transfer conductance varies with the 0.37 power of the air velocity for air flow normal to the liquid, with the 0.56 power for transverse flow and the 0.80 power for longitudinal flow. Each type of relative motion is present in the atmospheric tower.

A second phase of the project will be to establish the model laws of cooling tower behavior. Upon presentation of these laws, tests on models may be

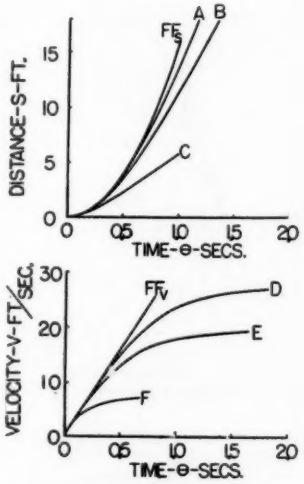

FIG. 16. DROP DYNAMICS. VELOCITY-DISTANCE-TIME CURVES FOR WATER DROPS FALLING FROM REST THROUGH STILL AIR

A, D — diameter = 5.0 mm
B, E — diameter = 2.0 mm
C, F — diameter = 0.5 mm
FF — free fall

extrapolated to the prototype with reasonable certainty. A third phase will be to attempt to formulate field testing technique and finally the fourth phase will be to study the economics of evaporative cooling.

DROP DYNAMICS

A study of the behavior of falling water drops will reveal results which may be applied to the problem of deck spacing in atmospheric (free convection) towers to the design of forced convection spray towers and to the design of

cooling ponds. The knowledge of the dynamical behavior of drops will serve as the introductory step to the study of the heat and mass transfer experienced by drops as upon passage through the tower.

This behavior is most readily summarized in the form of velocity-distance-time curves for various drop sizes. These curves are computed in the following manner:

1. *The equation of motion. The equation determining the path of the drop when falling downward is*

$$mg - C\frac{A\rho v^2}{2} = m\frac{dv}{d\theta}$$

where m = mass of drop, v = velocity of drop, θ = time, $\frac{dv}{d\theta}$ = acceleration of drop,

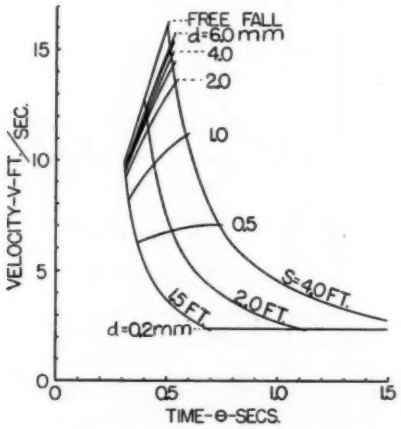

FIG. 17. DROP DYNAMICS. THE EFFECT OF THE DROP SIZE UPON THE TIME REQUIRED TO FALL DISTANCES OF 1.5, 2.0 AND 4.0 FT AND THE CORRESPONDING VELOCITIES ATTAINED. DROP FALLS FROM REST THROUGH STILL AIR

g = acceleration of gravity, ρ = mass density of the air (or any fluid through which the drop might fall), C = aerodynamic resistance coefficient defined by the equation $C = \frac{\text{Resistance offered by the air}}{\frac{1}{2}A\rho v^2}$, A = projected area of the drop normal to the direction of motion. Experimental data are available establishing C as a function of the Reynolds' Number for the drop, $Re = \frac{Vd}{\nu}$ where d = equivalent diameter of spherical drop of equal volume, ν = kinematic viscosity of the air.

A falling drop which encounters air resistance will be somewhat deformed compared to a true spherical shape. Photographs of falling drops have shown that a drop of diameter 1 or 2 mm or larger falling at the steady state velocity has the shape similar to that of an inverted acorn; *i.e.*, the leading surface is appreciably flattened while the trailing surface remains distinctly rounded but is slightly elongated.

This deformation gives rise to the question of how the projected area A and the equivalent diameter d should be measured. For want of a more satisfactory and convenient method, it has been the practice to refer these values to the equivalent spherical drop having the same volume as the actual drop. The satisfactory correla-

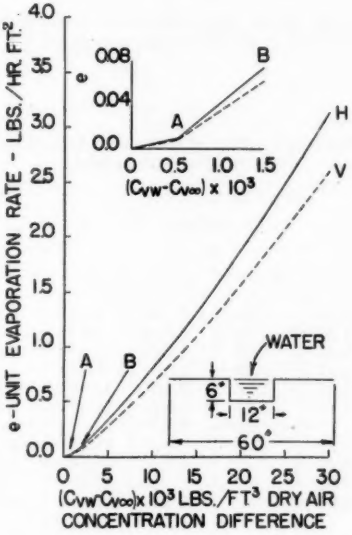

FIG. 18. UNIT EVAPORATION RATE FROM QUIET WATER SURFACES TO QUIET AIR AS A FUNCTION OF THE VAPOR CONCENTRATION DIFFERENCE AT THE LIQUID-AIR INTERFACE AND FAR AWAY

H — horizontal surface
V — vertical surface

EQUATIONS FOR H-SURFACE:

O — A	$e = 18.8\,\Delta c$	$O < \Delta c < 0.00052$
A — B	$e = -0.024 + 65\,\Delta c$	$0.00052 < \Delta c < 0.0015$
B →	$e = 251\,(\Delta c)^{\frac{1}{2}}$	$0.0015 < \Delta c < 0.030$

(Equations are applicable to air temperature of 70 F and relative humidities of 50 per cent.)

tion of the aerodynamic resistance coefficient as a function of the Reynolds' Number on this basis justifies this method.

2. Data on the steady-state velocity of falling drops and its application to the determination of the resistance coefficient as a function of Reynolds' Number.

Liznar,[7] who summarized the data of Schmidt[8] and Lenard,[9] gives magnitudes of

[7] Liznar, J., *Meteorol. Zeitschr.*, 31, 339 (1914).
[8] Schmidt, W., *Akad. Wiss. Wien Sitzungber.*, 118:2a, 71 (1909).
[9] Lenard, P., *Meteorol. Zeitschr.*, 21, 249 (1904).

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the steady-state velocity for falling water drops. Similar data are given by Flower.[10] Data on the increase of velocity over the steady-state velocity which is required to shatter a drop is reported by Lenard;[11] this information serves to establish the limiting velocity which is possible for a stable drop. For the air resistance of solid spheres, the data of Wieselsberger[12] is outstanding and has been re-checked by subsequent investigations of other workers.

The relation between the steady-state velocity and the drop diameter, revealed in Fig. 14, illustrates the manner in which the deformation of the larger drops gives rise

TABLE 1 — REPRESENTATIVE MAGNITUDES COMPUTED ACCORDING TO THE DATA OF LIZNAR

DROP DIAMETER	FREE FALL		d = 0.5 MM		d = 2.0 MM		d = 5.0 MM	
TIME SEC	v FT/SEC	s FT	v FT/SEC	s FT	v FT/SEC	s FT	v FT/SEC	s FT
0.1	3.18	0.16	2.85	0.145	3.15	0.15	3.18	0.16
0.2	....	0.64	4.75	0.538	6.05	0.61	6.32	0.638
0.3	....	1.45	5.84	1.078	8.82	1.37	9.94	1.43
0.4	....	2.57	6.40	1.69	11.10	2.36	12.12	2.50
0.5	16.08	4.02	6.73	2.36	12.95	3.57	14.62	3.84
0.6	....	5.79	6.94	3.04	14.47	4.93	16.91	5.42
0.68	....	....	7.02[b]	3.59[b]	....	....	....	....
0.7	....	7.88	....	....	15.66	6.45	19.02	7.23
0.8	25.73	10.29	....	4.43	16.59	8.08	20.85	9.22
0.9	....	13.02	....	....	17.30	9.82	22.26	11.37
1.0	....	16.08	....	5.85	17.81	11.54	23.40	13.64
1.18	....	....	....	....	....	....	....	18.00[a]
1.2	....	....	....	....	18.41	15.31	24.96	....
1.356	....	....	....	....	....	18.00[a]	....	....
1.4	....	....	....	....	18.84	....	25.82	....
1.6	....	....	....	....	19.15	....	26.45	....
1.66	....	....	....	....	19.20[b]	....	....	....
1.84	....	....	....	....	....	....	26.9[b]	....

[a] Limit of plot. Interpolation equations are used for greater distances.
[b] Steady state velocity.

to a greater increase in the resisting force of the air than the enlarged size increases the drop weight. According to the available data, it would appear that a drop of about 5 mm diameter falls with the maximum steady-state velocity. The extent to which the curve falls off for the larger drops is somewhat uncertain. Additional data are needed to completely establish the behavior of drops which have diameters of 6 and 7 mm.

Magnitudes of the limiting velocity for a stable drop, as determined by applying the velocity increments given by Lenard to the steady-state velocities are also given in the table of Fig. 14. These magnitudes may be utilized in the study of nozzle discharges and in determining the maximum possible size for a stable drop.

[10] Flower, W. D., *Proc. Phys. Soc. London*, 40, 167 (1928).
[11] Lenard, P., *Ann. Physik*, IV-65, 629 (1921).
[12] Wieselsberger, C., *Zeitschr. Flugtech. Motorluftschiffahrt*, 5, 140 (1914).

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To determine magnitudes of the resistance coefficient, C, the steady-state velocity and the condition $\frac{dv}{d\theta} = O$ is substituted into the equation of motion which is then solved for C. Results of such computations are given in Fig. 15.

The effect of the deformation of the drop in increasing the resistance above that for a sphere at the higher magnitudes of Reynolds' Number is also shown here, the resistance coefficient increasing to the magnitude corresponding to a flat circular plate (it may even go above this value, but the available data are of a limited range and do not establish this point). The effect of the surface tension of the drop upon the degree of deformation under given flow conditions may be obtained from the data of Flower for methyl salicylate, which possesses a surface tension about one-half of that for water. Magnitudes of Reynolds' Number corresponding to the steady-state and limiting velocities are also shown as they depend upon the drop size; it is seen that the two curves intersect at a drop diameter of about 8 mm, indicating that no larger drop could be stable. Measurements of the size of raindrops serve to confirm this point. In an ordinary heavy rain the largest drops are of the order of 5 to 6 mm in diameter. Heavy thunderstorms contain a few drops up to 6.5 or 7 mm in diameter.

The wide spread between the data of Flower and that of Liznar requires some explanation. At first it was thought that the divergence was due to a combination of different experimental methods and the effect of upstream turbulence, for Flower measured (photographically) the velocity of drops falling through still air, while Lenard measured the velocity of an air stream required to just balance the weight of the drop. However, subsequent computations yielding the distance-velocity curves indicate that Flower did not actually measure the steady-state velocity, for the computed distance of fall required to attain this velocity for the larger drops is about double the distance utilized in Flower's tests. It would seem that this difficulty arose because of Flower's use of a ballistic balance to determine the distance of fall necessary to give the steady-state velocity as manifested by a maximum in the throw of the balance upon the impact of a drop. But since the rate of change of velocity with distance becomes very small as the steady-state is approached, it is doubtful whether the true limit could be detected by a ballistic balance.

3. *Integration of the equation of motion.*

The form of the equation and the way in which C varies with Reynolds' Number make it necessary to resort to graphical integration. The velocity-time curve is obtained by evaluating

$$\int_0^\theta d\theta = \int_0^v \frac{dv}{g - C\frac{A\rho v^2}{2m}}$$

where the factor $\left(g - C\frac{A\rho v^2}{2m}\right)$ represents the acceleration per unit mass.

A second graphical integration yields the distance-time curve according to the relation

$$\int_0^s ds = \int_0^\theta v d\theta .$$

The velocity-distance-time curves have been obtained in this manner for drops of diameter 0.2, 0.5, 1, 2, 3, 4, 5, and 6 mm, both for the case in which the drop falls from rest and that in which it is discharged (either upward or downward) from a nozzle. Two sets of curves have been computed, one according to the data of Flower and one according to the data of Liznar. Even though it is believed that the magnitudes according to Liznar are the more accurate, the use of both sets of data will serve to indicate the effect of changes in the value of the resistance coefficient and the range of behavior which one may expect.

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These curves have been computed for the case in which the drop falls through still air, but they may be applied to the case in which the air is in motion upon recognition of the fact that the velocity of the drop as given is that relative to the air through which the drop is passing.

4. *Velocity-distance-time curves for drops falling from rest.*

Characteristic relationships can be brought out clearly for drops of 0.5, 2, and 5 mm diameter as representative magnitudes computed according to the *data of Liznar.* Magnitudes are tabulated in Table 1 and are plotted in Fig. 16.

5. *The time required to travel a given distance and the velocity attained in this distance as related to the diameter of a drop falling from rest.*

Data of this type have immediate application to the atmospheric cooling tower and are tabulated in Table 2 as well as being plotted in Fig. 17.

The data presented in Tables 1 and 2 on drop dynamics will serve as an introduction to a more comprehensive report which will be available within the year.

TABLE 2—DATA OF LIZNAR

DISTANCE OF FALL	S = 1.5 FT		S = 2.5 FT		S = 4.0 FT	
DROP DIAMETER MM	θ SEC	v FT/SEC	θ SEC	v FT/SEC	θ SEC	v FT/SEC
0.2	0.696	2.367	1.12	2.367	1.68	2.367
0.5	0.371	6.28	0.520	6.77	0.738	7.02
1.0	0.3275	8.10	0.439	9.80	0.582	11.12
2.0	0.315	9.18	0.412	11.34	0.533	13.50
3.0	0.311	9.50	0.403	11.87	0.516	14.17
4.0	0.310	9.56	0.402	12.00	0.514	14.66
5.0	0.309	9.59	0.401	12.15	0.512	14.90
6.0	0.308	9.60	0.400	12.16	0.510	15.05
Free fall	0.307	9.86	0.396	12.73	0.499	16.06

The following tasks are now being accomplished:

1. Correlation of the velocity-distance-time curves for drops discharged from a nozzle and decelerated by the air resistance.

2. Heat transfer computations for the evaporative cooling of falling drops.

3. Studies of the dispersion of jets and attempts to definitely evaluate the deformation of the drop as it encounters the resistance of the air.

4. Considerations relating to the dynamics of drop impacts and heat transfer accompanying such action.

EVAPORATION FROM A QUIET WATER SURFACE INTO QUIET AIR

These experiments [13, 14, 15] furnish the data from which the *minimum* rates of evaporation from a horizontal water surface may be predicted. No water surface existing in and about cooling towers will be bounded by quiet air. Mass and h at transfer will then occur in excess of that rate which is due only

[13] Sharpley, B. F., and Boelter, L. M. K., *Ind. Eng. Chem.*, 26, 1125-1131 (1938).
[14] Gordon, H. S. Thesis M. S. Degree 1938. Evaporation from a One-foot Diameter Water Surface into Quiet Air. (On file in the University of California Library, Berkeley.)
[15] Griffin, J. R. Thesis M. S. Degree 1939. Evaporation from a Horizontal Water Surface into Air by Free Convection. (On file in the University of California Library, Berkeley.)

to free convection of the air. Therefore the data presented in Fig. 18 may be considered as the minimum unit evaporation rate which will be experienced from water surfaces existing in cooling tower systems.

The data shown apply only for a 1-ft diameter pan, distilled water being held at the level of the surrounding 5 x 5 ft floor. The pan was completely housed in order to prevent cross flow due to thermal and dynamic wind forces.

The experimental data lie in a range in which the unit evaporation rates [pound per (square foot) (hour)] are not quite independent of pan diameter, varying approximately inversely as the quarter power of the diameter. As a

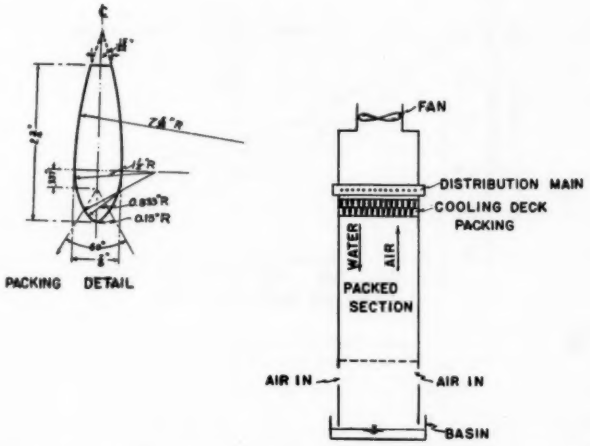

FIG. 19. CONSTRUCTION DETAILS OF AIRFOIL-SECTION PACKED TOWER

first approximation the results may be used for other surface areas with diverse surroundings and various wet- and dry-bulb temperatures of the air.

The analogy between heat transfer and mass transfer in free convection has been established for a limited range of the pertinent variables and for one geometrical arrangement. For heat transfer the conductance per unit area for the top of a horizontal surface is approximately 1.2 times that of a vertical surface.[16] Although based on meagre evidence and on the validity of the heat transfer—mass transfer analogy, the evaporation data obtained for a horizontal surface will be divided by 1.2 in order to obtain a first approximation to the mass transfer of vapor from a *stationary* vertical water sheet into quiet air at different humidities. These data are also shown in Fig. 18 and may be used to predict the minimum rate of evaporation from a vertical water sheet to quiet air.

[16] Bosch, M. ten, Die Wärmeübetragung, 3rd Ed., p. 167, Berlin, Julius Springer, (1936).

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RESULTS ON THE MECHANICALLY INDUCED DRAFT, CONTRA-FLOW, PACKED COOLING TOWER

A photograph of this tower is shown in Fig. 6 and construction details are revealed in Fig. 19. A description of the tower is included earlier in the paper. The piping arrangement is shown in Fig. 7.

The tower was tested[17] at various air rates and at various water rates as indicated in Fig. 20. Mass and heat balances were accomplished for each run.

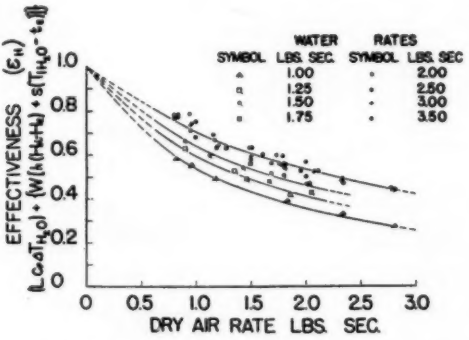

FIG. 20. THE EFFECTIVENESS OF THE AIRFOIL-SECTION PACKED TOWER AS A FUNCTION OF THE AIR RATE FOR VARIOUS WATER RATES

$$\varepsilon_H = 1 - 1.40e^{-\frac{f_D A}{G}\eta}$$

$\eta = 1.00$	$L \geq 1.75$ lb per second water rate
$\eta = 0.87$	$L = 1.50$ lb per second water rate
$\eta = 0.79$	$L = 1.25$ lb per second water rate
$\eta = 0.71$	$L = 1.00$ lb per second water rate

The average discrepancy was approximately ± 3.5 per cent. The pressure drop through the tower has not yet been measured. No attempt has been made to condition the entering air so that a complete series of runs for various humidities must be made throughout the period of a year (low humidities usually occur only in September). Entering air relative humidities ranged around 60 per cent, dry-bulb temperatures ranged from 65 F to 70 F, and entering water temperatures ranged from 80 F to 103 F.

The effectiveness of the tower was defined in the following manner:

ε_H = enthalpy reduction of water divided by the maximum possible enthalpy decrease of the water if the exit air is saturated at the inlet water temperature. This effec-

[17] London, A. L. Thesis M. S. Degree 1938. The Performance Characteristics of a Mechanically Induced Draft. Contra-Flow, Packed Cooling Tower. (On file in the University of California Library, Berkeley.)

tiveness ratio defines the tower performance as an air heater and humidifier and is shown as a function of the air rate in Fig. 20. It will be noted that the effectiveness is independent of the water rate down to 1.75 lb per second, below which partial wetting of the packing results.

A dimensionless ratio,[18] named the Number of Transfer Units, $\frac{f_D A}{G}$, *where* f_D = mass transfer conductance per unit area in pounds per hour foot^2 (pound/pound), A = area of packing in foot^2 and G = air rate in pounds per hour may be used as the generalized variable instead of the air rate G. The actual length of the tower (l) divided by the number of transfer units is called the height of a transfer unit, HTU. The HTU for the packed tower ranges from 3.5 to 8.5 ft. The behavior of the tower can be expressed empirically by the equation

$$\varepsilon_H = 1 - 1.40e^{-\frac{f_D A}{G}\eta}$$

where η = fraction of the packing which is wetted and e = Napierian base. $\eta = 1$ for water rates greater than 1.75 lb per second. Another effectiveness ratio which is based upon cooling performance may be defined as:

ε_c = enthalpy decrease of water divided by the maximum possible decrease, provided the water exit temperature coincides with the inlet air wet-bulb or

$$\varepsilon_c = \frac{\textit{decrease of water temperature}}{\textit{inlet water temperature—inlet air wet-bulb temperature.}}$$

The range of variation ε_H and ε_c is almost identical, although the variation is opposite. For instance, for the same liquid rate, the magnitude of ε_c for the high gas rate will correspond to that of ε_H for a low gas rate. The effectiveness as a cooling unit increases with increased gas rate.

A more complete analysis of the results is under consideration, which, when added to the pressure drop data, will be presented in a future paper.

CONCLUSIONS

1. Progress on the specific problem of evaporative cooling of water by direct contact with air is being made. The variables have been segregated and the effect of each independent variable will be studied separately. For instance, models of the full-scale atmospheric towers will be placed in a wind stream of known turbulence in order to study the air flow through them. Air-free water will be used to determine the effect of dissolved gases. Further progress will be made in the ensuing year.

2. The studies in drop dynamics and the related computations on the evaporative cooling of water drops to follow will make available design data of fundamental importance.

3. Several effectiveness ratios of cooling towers have been proposed based on the concept of an air heating device and of a water cooling device. Each has its place in design calculations.

ACKNOWLEDGMENTS

Technical and technological contributions were made by the following men, students, graduate students, faculty and mechanicians: T. F. Ascherfeld, H. E. Brown, V. H. Cherry, F. L. Dickey, H. L. Eagles, J. T. Gier, H. S. Gordon, J. R. Griffin, J. E. Gullberg, M. L. Hilleger, E. D. Howe, A. L. London, W. E. Mason, T. H. Morton, S. A. Murphy, Jr., H. B. Nottage, W. F. Pemberton, B. F. Raber, F. A. Ryder, H. J. Smith, and H. W. Sylvis.

[18] See reference 2, p. 490.

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The technical suggestions of the membership and committees of the Los Angeles and Golden Gate Chapters of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *Air Conditioning Society of San Francisco* serve to guide the work. The interest and activities in behalf of the project by Prof. B. M. Woods are stimulating and effective. Construction and maintenance of the equipment is accomplished through the aid of WPA Mechanical Engineering Project No. 8850.

The assistance of California industry is also gratefully acknowledged.

APPENDIX

LITERATURE ON COOLING TOWERS *

American Electrician: Oct. 1901
American Machinist: Feb. 6, Apr. 2, 1896
ASME Transactions: 1911; 1898
Canadian Engineer: June 15, 1926; Aug. 11, 1925
Cassier's Magazine: Apr. 1909
Concrete: Sept. 1930
Electrical Review (London): Sept. 13, 1912
Electrical Review (NY): Aug. 1920; Jan. 25, 1908; Feb. 16, 1907
Electrical World: Mar. 6, 1920; May 4, 1909
Electrician: June 1920; Oct. 1901
Engineering (London): Dec. 13, 1929; Nov. 14, 1919; Feb. 28, 1913; May 10, 1901
Engineering News: Dec. 5, Mar. 20, 1902; Sept. 27, 1901; July 13, 1899; July 8, 1897; Mar. 5, 1896
Engineering Record: Jan. 31, 1918; Sept. 26, 1910
Factory: Aug. 1925
Gas Age Record: Oct. 7, 1922
Gas Journal: Oct. 24, 1928
Glaser's Annalen: Feb. 1898; Oct. 1897
Ice and Refrigeration: Sept. 1912
Industrial and Chemical Engineering: May 1930
Industrial Engineering: Aug. 1915
Journal of Franklin Institute: Feb. 1911; June 1896
Mechanical Engineering: Oct. 1930; Aug. 1928; Nov. 1925; Feb. 1923; June, Aug. 1921; Feb. 1918; Feb. 1910; Nov. 1909
Petroleum Age: Apr. 15, 1925
Petroleum World: Oct. 1931; June 1930
Power: Apr. 1933; June 21, Sept., Oct. 1932; Sept. 29, July 21, May 19, Mar. 24, Feb. 3, 1931; Dec. 9, Dec. 2, Nov. 25, Aug. 12, Aug. 5, Apr. 15, Mar. 1, 1930; Dec. 24, Dec. 3, Oct. 8, 1929; June 5, May 1, Mar. 20, 1928; Dec. 7, Oct. 12, May 4, 1926; Nov. 17, 1925; Jan. 15, 1924; Feb. 20, 1923; Nov. 8, 1921; Dec. 2, June 10, June 3, May 13, 1919; June 25, Mar. 26, 1918; Jan. 26, 1915; Aug. 18, 1914; Nov. 11, Oct. 7, Mar. 11, 1913; Oct. 4, Aug. 30, 1910; June 22, Feb. 9, Jan. 12, 1919; July 28, July 14, 1908; Oct. 1901; Nov. 1898
Power Plant Engineering: Apr., Sept. 1932; Aug. 1, 1929; Oct. 15, Sept. 15, 1927
Proceedings of South African Engineering: Sept. 1909; Feb. 1908
Refrigerating Engineering: May 1932; Oct. 1931; Dec. 1924; Dec. 1923; Aug. 1922
Refrigerating World: Oct. 1931; Sept. 1929; Jan. 1928; Mar., Sept. 1923
Structural Engineer: Mar. 1930
The Engineer (London): May 30, 1919; Apr. 23, 1909
The Engineer (USA): July 15, 1903
The Oil and Gas Journal: Sept. 29, 1932; Aug. 14, 1930; June 28, 1928

* Compiled by Wm. H. Marchant for a thesis presented for the degree of Engineer at Leland Stanford University, June 1934.

No. 1139

FALL MEETING, OCTOBER, 1939

ATLANTA was the scene of the Fall Meeting of the Society, held October 30-31 at the Atlanta Biltmore Hotel. Members from 22 states and Canada made up the total registration of 180 who attended the three technical sessions and enjoyed the social events, arranged by the Atlanta Chapter. Pres. J. F. McIntire called the first Fall Meeting into session at 9:30 a. m., Monday, October 30, and C. L. Templin welcomed the visiting members, guests and ladies.

Prof. G. L. Tuve, Cleveland, presented his paper entitled The Use of Air-Velocity Meters by Professor Tuve, D. K. Wright, Jr., and L. J. Seigel, outlining the results of research sponsored by the ASHVE in cooperation with the Case School of Applied Science. The paper reported on three years of investigation covering methods of instrument calibration and a study of the practical use of instruments for measuring air stream characteristics and air quantities. The authors emphasized the point that accurate calibrations are necessary and said that calibration methods for all instruments should be standardized. The paper was illustrated by lantern slides and proved to be of exceptional interest to the members present.

Prof. A. P. Kratz presented written discussions prepared by M. K. Fahnestock and A. E. Hershey of the research staff of the University of Illinois. The various questions asked were answered by D. K. Wright, Jr., one of the authors. The written comments of Prof. D. W. Nelson, University of Wisconsin, and C. H. Randolph, Milwaukee, were read by the secretary of the Society, A. V. Hutchinson. S. H. Downs, Kalamazoo, and J. L. Blackshaw, Atlanta, presented oral discussions, and the final closure for the authors was made by Mr. Wright.

A discussion of thermostatic performance was given by B. E. Shaw, Goshen, Ind., in the paper entitled A Study of Certain Factors That Affect the Droop Characteristics of Thermostats by B. E. Shaw and La Verne Lyon, which was illustrated by slides of tabulated test data and diagrams of apparatus used in the various tests reported. The authors presented five specific conclusions with reference to the cause of droop and offered the suggestion that further investigation was essential.

The afternoon program consisted of four papers. President McIntire introduced C. T. Baker, vice-chairman of the Committee on Arrangements, who

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presented his paper Food Preservation in the South (complete paper published in December, 1939, ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*), wherein he describes methods employed in Georgia and the Carolinas for the use of air conditioning in the preservation and storage of perishable crops. He also describes the design of several plants for maintaining correct temperatures and humidities. Those taking part in the oral discussion were Messrs. Carter and Boester, St. Louis, Oaks and Fleisher, New York, Lewis, Cincinnati, and C. P. Lichty, Birmingham, Ala.

The subject of application of storage refrigeration was discussed by Carl F. Boester in his paper entitled The Application of Storage Refrigeration to Air Conditioning, which describes the design and operating characteristics of installations using ice accumulating evaporators. Mr. Boester gave a careful analysis of the usage factor and also gave some data on operating costs. In the discussion questions were asked by C. T. Baker, John Carter, F. J. Dean Jr., F. B. Herty, and W. L. Fleisher.

F. C. Houghten, Director of the Society's Research Laboratory, presented the paper entitled A Standard Air Filter Test Dust by Frank B. Rowley and Richard C. Jordan. This paper resulting from the cooperative research sponsored by the ASHVE and the Engineering Experiment Station of the University of Minnesota describes the fundamental requirements of dust used for filter work, and in the conclusion points out that dusts are of two general classes insofar as they affect air filter ratings. Some oral comments were given by W. L. Fleisher, chairman of the Society's Committee on Research and a prepared discussion was presented by Arthur Nutting, Louisville, Ky.

The final paper in the afternoon session was given by John H. Carter, St. Louis, Mo., in his paper entitled An Analysis of the Air Conditioning Contractor's Business Relationships (complete paper published in November, 1939, ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*). The author covered the subject of customer's relations, consultant's role, the contractor's and distributor's obligations in the field of air conditioning and pointed out that there must be a proper balance between the business and the technical sides for a contractor to operate successfully.

Get-Together Luncheon

On Monday at 12:30 p. m. a get-together luncheon for the members and ladies was held in the main dining room of the Atlanta Biltmore Hotel, and C. L. Templin, president of the Atlanta Chapter, was the master of ceremonies. The guests at the head table were introduced after which Dr. M. L. Brittain, President of Georgia Tech, addressed the group on Technology and Southern Industry. In his remarks Dr. Brittain pointed out that there was a fertile field for the type of research conducted by the Society in the South and offered the facilities of Georgia Tech at any time for cooperative research investigation.

At the conclusion of the afternoon technical session members and ladies gathered for the Friendship Hour and enjoyed a musical program and appropriate refreshments.

Banquet

The outstanding social event of the meeting was the dinner and dance held in the ballroom of the Atlanta Biltmore on Monday evening. Oliver J. Prentice acted as toastmaster and introduced the Society Officers and Council, and pointed out that 14 of the 17 men were present for this meeting. He noted that in the audience were W. H. Driscoll, past president of the Society, W. J. Olvany, president of the *Heating, Piping and Air Conditioning Contractors' National Association,* and the presidents of several chapters, H. E. Sproull, Cincinnati, O. O. Oaks, New York, L. P. Saunders, Western New York, Tom Brown, Chicago, Prof. R. B. Rice, Raleigh, and C. L. Templin, Atlanta.

President's Address

Pres. J. F. McIntire was called upon for a brief talk and said:

"Professional engineering societies depend largely on research done in private laboratories for advancement. The ASHVE is the exception—20 years ago we established our own Laboratory at Pittsburgh in connection with the Bureau of Mines.

"The purpose was to have a laboratory dedicated to the conduct of fundamental research. Commercial testing was barred. An important function of the Laboratory has been to supply basic data for the development of codes and standards.

"The Government furnishes, without charge, space, light, heat, technical assistance and shop service, thus increasing the results per dollar of our expenditures.

"The scope of the research activities of the Society is also greatly extended by means of cooperative agreements with such universities as California, Illinois, Minnesota, Pennsylvania, Pittsburgh, Wisconsin, Lehigh, Princeton. Agricultural and Mechanical College of Texas, Case School of Applied Science, Michigan College of Mining and Engineering, Massachusetts Institute of Technology and U. S. Navy.

"So you can see why we were so interested at our luncheon today when President Brittain suggested a cooperative agreement between Georgia Tech and the Society.

"Our Chairman of Research, W. L. Fleisher, vice-chairman J. H. Walker and the Director of our Research Laboratory, F. C. Houghten, are in attendance at this meeting and you can be sure that they will not leave Atlanta without following up Dr. Brittain's suggestion. Cooperative agreements avoid duplication of effort and coordinated activities to produce maximum results per dollar of expenditure. Many problems need investigation in all three fields of air conditioning, industrial, commercial and comfort. This community is interested in all of these."

The guest of honor for the occasion was Robert T. Jones, well known Atlanta attorney and famous golfer, who graduated from Georgia Tech as a mechanical engineer and found that there was a lot of engineering in golf, but that golf practice seriously curtailed the practice of engineering. He recalled some of his early experiences in a rather brief engineering career.

The speaker of the evening was O. B. Keeler, well known writer and commentator on sports, who is often referred to as "Bobby Jones Boswell." He delighted the audience of 225 with some amusing stories and gave a comprehensive review of interesting events in the sports world, and described some of the title matches played by Bobby Jones on many golf courses throughout the world, and also turned back some of the pages of baseball, football, tennis and boxing history.

During dinner instrumental music was played by Forrest Traylor and Forrest

Traylor, Jr., and a novelty number by Bobby Staples. At the conclusion of dinner Harry Hearn and his orchestra played music for dancing.

On Tuesday morning the final technical session was called to order by Pres. J. F. McIntire, who introduced F. B. Lanham, research agricultural engineer, University of Georgia, who presented his paper on Agriculture Has Engineering Problems in Heating and Ventilating (complete paper published in December, 1939, ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*). Of particular interest were his descriptions of artificial drying of hay and the curing and storing of sweet potatoes.

Lester S. O'Bannon, research engineer, University of Kentucky, Lexington, presented his paper dealing with A Laboratory for Use in Determining the Influence of Various Temperatures and Relative Humidities on the Final Quality of Air-Cured Tobacco (complete paper published in November, 1939, ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*). This paper reported on some original research work and included a detailed description of the equipment and controls used to determine desirable temperatures and humidities for tobacco curing.

The concluding paper of this session was given by Paul D. Close, Chicago, in his paper entitled Permissible Relative Humidities in Humidified Buildings (complete paper published in December, 1939, ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*). The author gave valuable data on the subject of condensation which occurs on the interior surfaces of buildings whenever such surfaces are below dew-point temperatures. He presented an ingenious chart for quickly determining the point at which condensation occurs on walls, roofs or glass under various atmospheric conditions.

Resolutions

First Vice-President Giesecke inquired whether there was any unfinished business and hearing no response, asked if there was any new business to be brought before the meeting, and Merrill Blankin, John Everetts, Jr. and G. L. Wiggs, Montreal, offered the following resolutions which were unanimously adopted:

Whereas, this concludes not only the first Fall Meeting of the Society ever held and the first meeting held in the deep South, but it concludes one of the most successful and delightful meetings of the Society.

We wish to express and record the hearty appreciation and the sincere thanks of the Society.

To the Officers of the Atlanta Chapter and particularly its Committee on Arrangements for the splendid program and their untiring efforts in making this meeting a success.

To the local newspapers and trade publications for the cooperation and publicity.

To the Atlanta Biltmore Hotel for their fine accommodations and efficient service.

To the Atlanta Convention and Visitors Bureau for their great help and for arranging to broadcast Mr. Kent's speech over radio station WSB.

To the transportation companies for their excellent service.

To the various speakers for the time and effort they devoted to the preparation and presentation of their papers.

And lastly, but not least, to the ladies, without whose presence the meeting would not have been complete, and who so graciously upheld the high traditions of the South.

The ladies committee was very active during the two-day meeting and provided sightseeing and shopping trips for the visiting ladies. All enjoyed

a breakfast-bridge on Tuesday morning. The sightseeing trips included the residential section, Wren's Nest, the home of Joel Chandler Harris, the author of Uncle Remus stories, and the home of Margaret Mitchell of "Gone With the Wind" fame. A group of 25 visited the Cyclorama depicting the battle of Atlanta. At the breakfast-bridge 22 ladies were present and table prizes were awarded to Mmes. J. F. McIntire, F. E. Giesecke, M. C. Beman, Adams, and Miss Dorothy E. Gulbransen. Each visiting lady was presented with an attractive loose powder compact souvenir.

On Tuesday afternoon some of the members visited Georgia Tech, the Exposition Cotton Mills, Techwood Housing Project, and the Coca-Cola bottling plant, while several foursomes toured the East Lake Country Club course to test their skill as golfers.

Many of the members arrived in Atlanta on Saturday morning, October 28, and the Reception Committee was on hand to greet them at the Atlanta Biltmore. A group of 16 attended the football game and saw Georgia Tech beat Auburn 7 to 6. Others went to the East Lake Country Club for a round of golf.

Council

On October 29, at 10:00 a m. President McIntire opened the Council meeting attended by 14 members.

During the afternoon the Council members enjoyed a sightseeing trip through Atlanta, and at 4:00 p. m. a group of 30 members and ladies attended the Song Service of negro spirituals given by the Big Bethel Choir of 75 voices in the Bethel A. M. E. Church.

PROGRAM FALL MEETING

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
ATLANTA BILTMORE HOTEL, ATLANTA, GA. OCTOBER 30-31, 1939

October 28

1:00 P.M. Reception of early arrivals.
2:00 P.M. Football Game—Georgia Tech vs. Auburn (Alabama Polytechnic Institute).

October 29

10:00 A.M. Informal Golf Matches—East Lake Country Club.
11:00 A.M. Council Meeting.

October 30

8:30 A.M. Registration.
9:30 A.M. *Technical Session*—
The Use of Air Velocity Meters, by Prof. G. L. Tuve, D. K. Wright, Jr., and L. J. Seigel.
A Study of Certain Factors that Affect the Droop Characteristics of Thermostats, by B. E. Shaw and La Verne Lyon.
10:00 A.M. Ladies Shopping Trip.
12:15 P.M. Get-together Luncheon (Members and Ladies).
Speaker—Dr. M. L. Brittain, President Georgia Tech—Subject: Technology and Southern Industry.
1:45 P.M. Ladies Sightseeing Trip.

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2:00 P.M. *Technical Session*—
Food Preservation in the South, by C. T. Baker.
Application of Storage Refrigeration to Air Conditioning, by C. F. Boester.
A Standard Air Filter Test Dust, by Prof. F. B. Rowley.
An Analysis of the Air Conditioning Contractor's Business Relationships, by John H. Carter.

4:00 P.M. Friendship Hour for Members and Ladies (Biltmore Terrace). Music by Eleanor Hodges String Trio.

7:00 P.M. Autumn Dinner and Dance (Dinner Music by Forrest Traylor).
Guest of Honor—Robert T. Jones—Atlanta Attorney and Golfer extraordinary.
Speaker—O. B. Keeler—Sports Commentator and Member of Collier's All American Board.

October 31

9:30 A.M. *Technical Session*
Agriculture has Problems in Heating and Ventilating, by Prof. F. B. Lanham.
A Laboratory for Use in Determining the Influence of Various Temperatures and Relative Humidities on the Final Quality of Air-Cured Tobacco, by Prof. L. S. O'Bannon.
Maximum Permissible Humidities in Humidified Buildings, by Paul D. Close.

10:00 A.M. Ladies Breakfast-Bridge.

1:30 P.M. Golf Tournament: East Lake Country Club (Green Fees $2.00).

1:30 P.M. Inspection Trips—(1) Exposition Cotton Mills; (2) Techwood Housing Project; (3) Residential Tour and Coca-Cola Plant.

2:00 P.M. Golf Tournament—East Lake Country Club.

COMMITTEE ON ARRANGEMENTS

LAURENCE F. KENT, *General Chairman.*

C. T. BAKER, *Vice-Chairman.*

Banquet: E. W. KLEIN, *Chairman;* W. P. SHERMAN, LEO SUDDERTH, JR., J. W. PARKER.

Finance: C. B. COLE, *Chairman;* R. F. HAHN, L. F. LAWRENCE, JR., R. E. NOLAN, C. S. MINOR, JR., LEWIS A. HAMILTON.

Ladies: T. T. TUCKER, *Chairman;* E. R. FOSS, SPOTSWOOD PARKER.

Publicity: M. M. CROUT, *Chairman.*

Reception: C. L. TEMPLIN, *Chairman;* W. J. MCKINNEY, F. L. LASETER, BLACKMAN DUNN, R. E. VERNON, CARL JOHNSON.

Registration: S. W. BOYD, *Chairman;* L. L. BARNES, M. G. DRISCOLL, R. B. STOTZ.

Sports: A. H. KOCH, *Chairman;* G. H. BRODNAX, JR., G. T. GUNNELL, C. L. SCHROEDER, JR.

Transportation and Tours: C. E. BROCKINTON, *Chairman;* L. L. HAMILTON, R. D. KELLY, T. R. YARBOROUGH, L. H. HOBSON.

No. 1140

THE USE OF AIR-VELOCITY METERS

By G. L. TUVE,* D. K. WRIGHT, JR.** AND L. J. SEIGEL,† (*MEMBERS*),
CLEVELAND, OHIO

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with Case School of Applied Science

SUMMARY AND CONCLUSIONS

RESEARCH work on air-velocity instruments conducted at Case School of Applied Science in cooperation with the Committee on Research of the ASHVE has been in progress for about three years and has proceeded along two main lines: (1) Investigation of methods for instrument calibration; (2) A study of the practical use of instruments for measuring air-stream characteristics and air quantities.

In this report, a comparison is made of several methods of instrument calibration, and calibration curves for typical instruments are shown. Use of the various instruments is also reported, applying to supply openings, air intakes and free air streams. For the free air streams an analysis is based on the law of conservation of energy.

Several instruments are available for air velocity measurement, and calibration curves indicate that the instruments themselves give highly consistent readings. Each instrument has its special advantages, the rotating-vane anemometer being the best averaging instrument, and the bridled-vane anemometer (Velometer) the most convenient to use for *spot readings*. The heated thermometer has the widest velocity range, the heated thermocouple is best adapted for distant reading, and the Kata-thermometer is the least expensive.

Accurate calibrations are quite necessary, and calibration methods for all instruments should be standardized. Instruments may be calibrated in a wind-tunnel air stream having a working section of uniform velocity, located at the discharge of a rounded-entrance nozzle or contraction, but calibration in other types of air streams is not satisfactory because other velocity patterns cannot

* Professor of Heat-Power Engineering, Case School of Applied Science.
** Instructor in Mechanical Engineering, Case School of Applied Science.
† Cleveland Electric Illuminating Co., Cleveland, Ohio.

Presented at the Fall Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Atlanta, Ga., October, 1939.

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be duplicated. If a rotating arm is used for calibrations, its length should be at least 10 ft to obtain accuracy within 3 per cent.

For volume measurements at intake or discharge openings, the rotating-vane anemometer is the most convenient, since a traverse may be made and the average velocity obtained in one reading, but a correction factor of 0.70 to

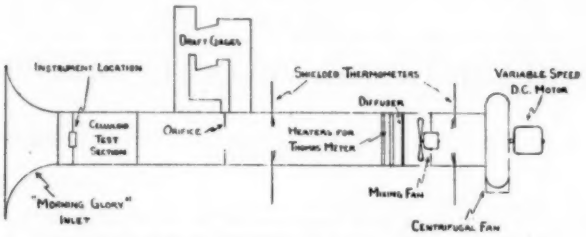
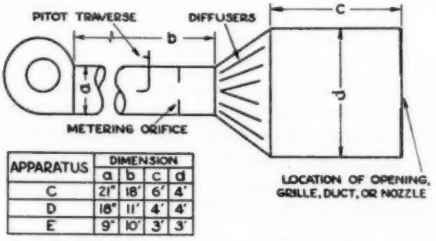


FIG. 1. 12-IN. CALIBRATING TUNNEL—APPARATUS *B*



APPARATUS	DIMENSION			
	a	b	c	d
C	21"	18'	6'	4'
D	18"	11'	4'	4'
E	9"	10'	3'	3'

FIG. 2. PLENUM BOX TEST UNITS

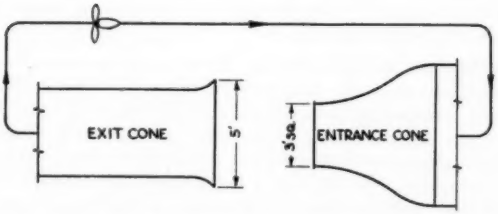


FIG. 3. TEST SECTION OF GÖTTINGEN-TYPE TUNNEL—APPARATUS *F*

1.05 is usually necessary to compensate for the specific conditions of use. Correction factors for various conditions are given herewith, but a further comprehensive investigation of the use of the anemometer is urgently recommended. It should be possible to develop a set of simple rules and correction factors which will give engineers confidence in this instrument even for measurements where an accuracy within 2 per cent is required.

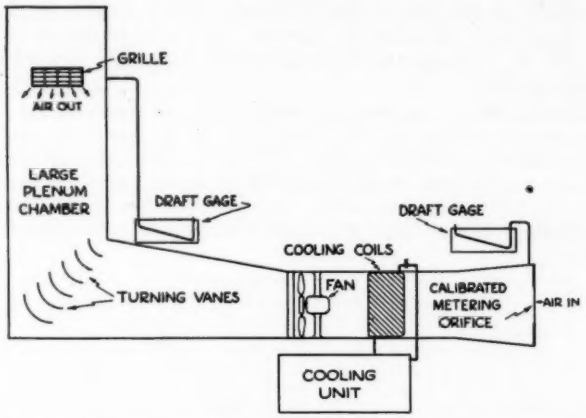


FIG. 4. TEST UNIT FOR GRILLES AND REGISTERS—APPARATUS *G*

For measurements of air velocities in a free air stream in a room, the Velometer is the most convenient instrument because of the large number of readings that can be obtained in a short time. The heated-thermometer and hot-wire instruments have possibilities for low velocity work, especially in the laboratory, and further investigations should be made in their application.

SCOPE OF AIR-MEASUREMENT PROBLEM

Every air conditioning problem involves the moving of the air *to* the conditioned space and the distribution of air *within* the conditioned space. The

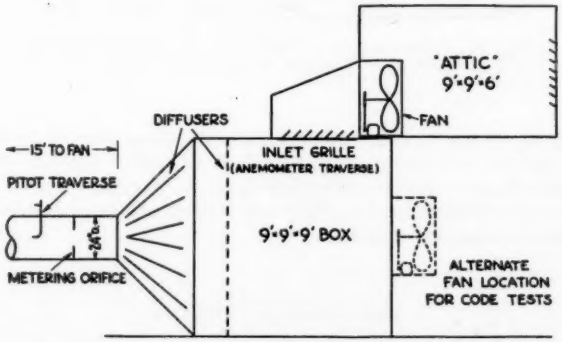


FIG. 5. TEST UNIT FOR LARGE PROPELLER FANS—APPARATUS *H*

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public evaluates an air conditioning installation almost entirely by the quality of the air distribution, and unless the engineer can make satisfactory air velocity measurements he is risking public dissatisfaction with what may otherwise be an excellent system.

In ventilating and air conditioning work, there are four common occasions for measuring air velocity: (1) In a duct; (2) At the face of a supply outlet; (3) At an intake or suction opening; (4) In an open space or room. Duct velocities are usually high enough to permit of direct impact measurement by the Pitot tube and manometer, while room air velocities range down to the barely perceptible air movement associated with a *draft*. The measuring of an

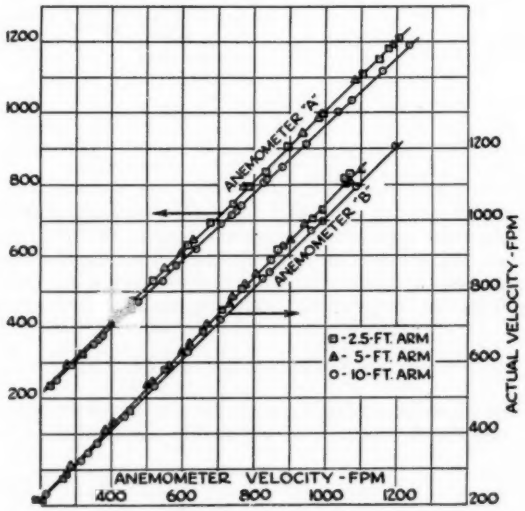

FIG. 6. ANEMOMETER CALIBRATION BY APPARATUS *A*, EFFECT OF LENGTH OF ROTATING ARM

air velocity of 50 fpm is quite different from the measuring of 1000 fpm, and calls for different instrument characteristics. A second problem arises from the fact that the air velocities in various parts of a stream are different, and this *velocity distribution* greatly affects the readings of large instruments such as the common 4-in. anemometer.

MEASURING AIR VELOCITY

In ventilating and air conditioning practice, the measurements of air velocity are usually in the range below 2000 fpm, and the present discussion will be

confined to this range. The three purposes of air-velocity measurement in air conditioning work are: (1) For sound-level or noise control; (2) For securing proper air distribution; (3) In order to avoid *drafts.* To satisfy this last requirement, measurements must be made of velocities well below 100 fpm.

In view of these several requirements, the engineer would like further information on the following questions:

1. What are the advantages and what are the limitations of each of the common air-velocity meters?
2. How may air-velocity meters be calibrated, and how accurate are the calibrations?
3. What special precautions are necessary for using instruments in air streams:

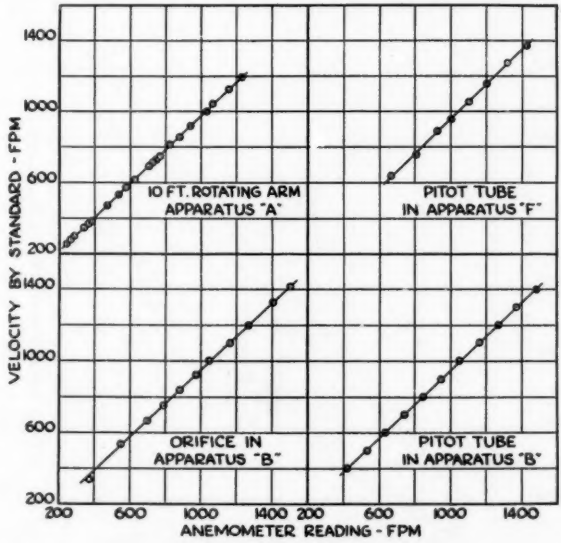

FIG. 7. CALIBRATION OF ROTATING-VANE ANEMOMETER *A* BY FOUR METHODS

(a) Discharging from supply openings? (b) Entering intake openings? (c) Traversing space in an open room?

The advantages and limitations of the various instruments become apparent when the methods of calibration and methods of application of the instruments are investigated, hence the problem has been attacked from that angle.

EXPERIMENTAL WORK

Some of the equipment which was made available for this work is illustrated by the diagrams of Figs. 1 to 5. Fig. 1 is a diagram of a 12-in. diameter

wind tunnel of the draw-through type with a large bell-mouth entrance and a transparent working section. This unit was used for instrument calibrations, and the velocity across its working section was strictly uniform as far as could be determined with a Pitot tube (except within 0.5 in. of the wall). Three measuring standards were used in this tunnel: (1) A thin-plate orifice; (2) An electric-thermal meter; (3) A Pitot tube. All three of these primary standards agreed very closely (within 1 per cent at the higher velocities).

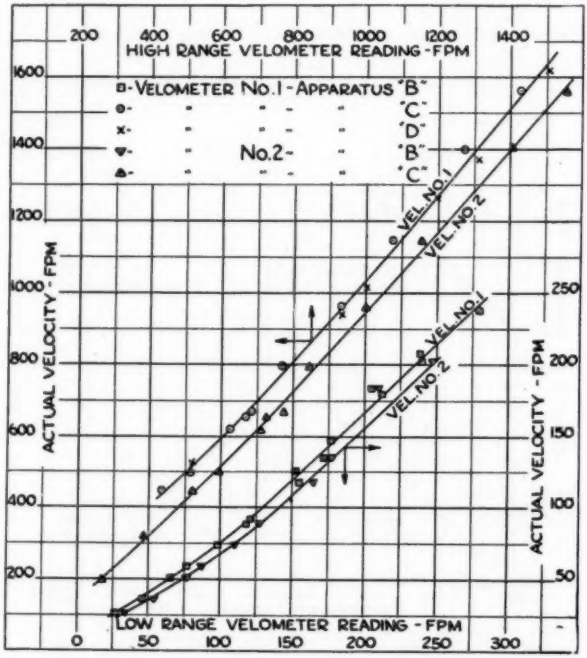

FIG. 8. VELOMETER CALIBRATIONS

Fig. 2 gives the dimensions of three different metering tunnels of the blow-through type, each with a diffuser and a plenum chamber or still-box. Fig. 3 is a diagram of a Göttingen-type recirculating wind tunnel with a 36-in. square working section, located in the Case Aerodynamics Laboratory. Fig. 4 shows a large plenum-box unit used mainly for the study of supply outlets and grilles. Additional inlet or discharge ducts were attached as required when using the units designated as Apparatus *C*, *D*, *E* and *G* for studies of supply and exhaust grilles. Apparatus *H* of Fig. 5 is a large set-up for comparison of the methods of measuring the air handled by large propeller fans, and by typical attic fan installations.

In addition to the wind-tunnel units, a rotating-arm apparatus (called hereafter Apparatus *A*) was used for anemometer calibrations. Interchangeable rotating arms of 2½ ft, 5 ft and 10 ft lengths were provided, and the unit was equipped with a solenoid starter and electric timer, operated from the same switch.

These eight test units were all used in the experimental work reported herein, and they are identified by letter in the tables and figures which are included. One of the chief methods for insuring accuracy in this work was to repeat the tests on two or more of these pieces of equipment, using different observers.

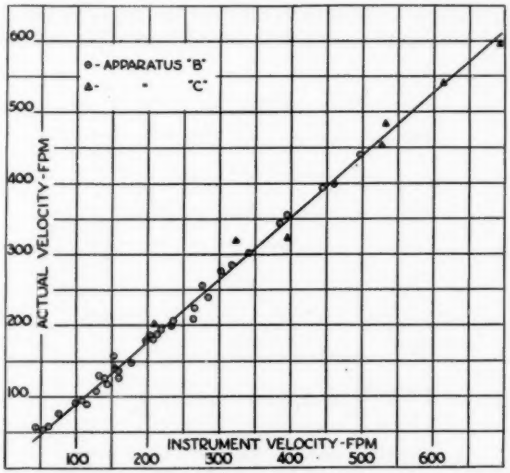

FIG. 9. HEATED THERMOMETER ANEMOMETER CALIBRATION

In the use of this test equipment, all measurements of Pitot tube, orifice, or plenum pressures were made with special inclined manometers which could be read to 0.001 in. water, and which were calibrated to this accuracy over their entire range by comparison with a sensitive micro-manometer. Special precautions were taken to prevent accidental leakage from ductwork and plenum chambers, and leakage calibrations were made in all cases.

CALIBRATION METHODS

Inquiry among the manufacturers indicates that they are using a wide variety of calibration methods. Many instruments are being calibrated by the secondary method of checking one instrument against another one of similar design which has in turn been calibrated by some primary method. One manufacturer indicated that he was using as a standard an instrument which had been calibrated

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in 1918. In several cases the air stream discharged from a centrifugal fan through an ordinary open-end round duct is being used for the comparison.

There are two classes of calibration methods. One is to move the instrument in still air, the other is to move the air and to fix the instrument in the air stream, comparing its reading with that of another velocity instrument or with a volume measurement by the equation $Q = AV$. The large amount of space and the complicated apparatus required for the straight-line towing of an instrument makes this method generally impractical. Swinging the instrument on a rotating arm is the next best substitute, and for anemometer calibrations, the rotating-arm apparatus is available both at the U. S. Bureau of Standards and at the National Physical Laboratory, London.

An investigation was undertaken in the Case laboratory to ascertain the effect of the length of rotating arm when this method is used for calibrating rotating-

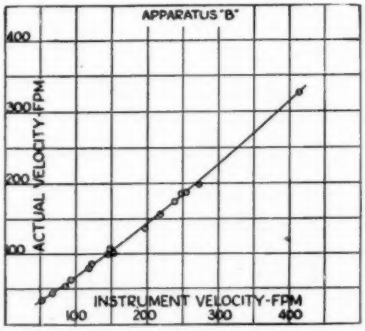

FIG. 10. HEATED THERMOCOUPLE ANEMOMETER CALIBRATION

vane anemometers. The results for two typical 4-in. anemometers are shown in Fig. 6. Three arm-lengths were used, 30-in., 60-in., and 120-in., and both the instrument control and the timing were accomplished by electrical methods. It will be noted that the two shorter lengths of arm gave almost identical results, but the results with the 10-ft arm agreed much more closely with the wind-tunnel method of calibration, as shown in Fig. 7. If the average of the three wind-tunnel calibrations is taken as the standard, an anemometer reading corrected by the calibration with a 10-ft rotating arm gives a result which is about 3 per cent high, and the shorter rotating arms give a result which is about 7 per cent high. The instrument mounting as well as the rotating arm itself must be of such design as to produce a minimum of air disturbance.

When an instrument is calibrated in a moving air stream, the instrument is more easily read. In fact it would be almost impossible to read some instruments on a rotating arm (the Kata-thermometer, for instance). But the moving air stream involves two difficulties. First, a stream of uniform velocity across a large section must be provided, and, second, some primary method of measuring the air-stream velocity must be used as a standard. A

third requirement for certain instruments is that the static pressure within the working section must be strictly uniform. Research on aerodynamic wind tunnels has proven that the first and last requirements may be met only when the working section is located at or just beyond the throat of a rounded-entrance nozzle with a large reduction ratio (see Figs. 1 and 3).

Given a uniform velocity across the working section, several standard methods of velocity measurement are available. In this investigation the orifice, the Pitot tube and the electric-thermal meter were used and the results agreed closely by the three methods. Calibrations of various instruments by the wind-tunnel method, using a nozzle-throat as the working section in all cases, are shown in Figs. 7 to 11.

The consistency of measurements with the rotating-vane anemometer is well illustrated by Fig. 7. It is also concluded that since the same methods were

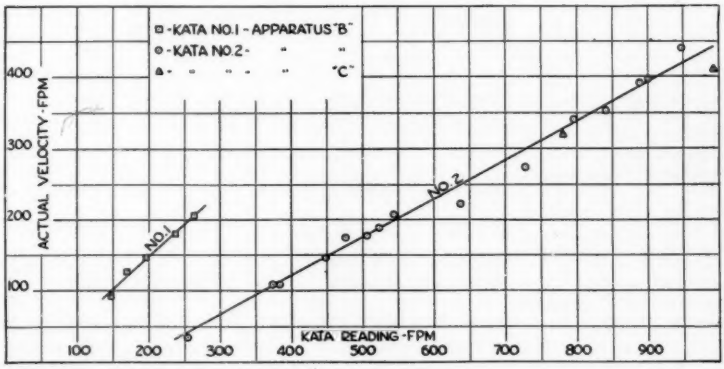

FIG. 11. KATA-THERMOMETER CALIBRATIONS

used throughout, the *scattering* of points in Figs. 8, 9 and 11 is due to the instruments and their manipulation, rather than to the calibration methods, since this scattering does not appear in Fig. 7, and practically all observed points were plotted in every case.

The low-range Velometer calibrations are made with the instrument box only, and the high-range calibrations with tube and *jet*. The jet of Velometer No. 1 was a ¼-in. round opening, that of No. 2 was an 0.05-in. x 4-in. slot. A straight-line calibration curve was obtained for the rotating-vane and the heated thermometer anemometers, but the Velometers and the heated thermocouple definitely require curves which are concave upward. The wide-range capacity of the heated thermometer is an outstanding feature, and the same principles could be applied to the heated thermocouple, although the one here calibrated was strictly a low-velocity instrument (Hukill design[1]).

[1] An Anemometer for Measuring Low Air Velocities, by W. V. Hukill. (*Refrigerating Engineering*, October, 1934, p. 197).

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Kata-thermometer No. 1 (Fig. 11) was unfortunately broken after partial calibration, and a duplicate thermometer (No. 2) purchased from the same maker showed entirely different characteristics.

AIR MEASUREMENTS AT DISCHARGE OUTLETS

The character of the air flow at an outlet depends mainly upon the approach conditions immediately upstream, and upon whether the outlet is free-open or

TABLE 1—ANEMOMETER TRAVERSE DATA FOR AIR OUTLETS AND SUPPLY GRILLES

TYPE OF GRILLE	TYPE OF APPROACH	VELOCITY RANGE, FPM	SIZE, IN.	PER CENT FREE AREA	NUMBER OF TESTS	ANEMOMETER FACTOR, F[a]	
						Based on Free Area	Based on Effective Area
Free-open, no grille	Plenum	300–1600	18 x 6	100	5	0.67	0.67
Free-open, no grille	Plenum	700–1700	12 x 6	100	5	0.73	0.73
Steel-bar grille, H bars on ½-in. centers	Plenum	250–500	12 x 6	78	4	1.07	0.95
Steel-bar grille, V and H bars	Plenum	600–1000	18 x 6	79	4	1.03	0.92
Steel-bar grille, H and V bars	Plenum	250–800	20 x 8	79	9	1.02	0.91
Steel-bar grille, H and V bars, fine mesh	Plenum	400–1400	24 x 8	76	9	1.01	0.88
Free-open, no grille	Small duct	300–900	12 x 6	100	11	1.06	1.06
Steel-bar grille, H bars on ½-in. centers	Small duct	300–1300	12 x 6	78	4	1.14	1.01

[a] F is the anemometer correction factor, defined by the equation: Actual cfm $= FAV$, *where* A is the area and V is the average velocity by anemometer traverse.

covered with a grille or register. There are, of course, infinite varieties of directional-flow outlets, but it is important to understand the straight-flow outlet first, and this discussion will be confined to the latter. There are two extreme types of locations for straight-flow outlets: (1) In the wall of a plenum chamber; (2) At the end of a duct the same size as the outlet.

When air is discharged from a thin-walled plenum chamber or a very large duct, through an ordinary small square-edged outlet, either free-open or covered by a low-resistance grille, there is a jet contraction similar to that obtained with a round, sharp-edged orifice in a thin plate. In addition, the static-to-velocity pressure conversion continues beyond the plane of the opening. These two facts make it very difficult (if not impossible) to obtain a true *average* velocity across the face of such an opening for the accurate application of the continuity equation, $Q = AV$. As a practical solution for this difficulty the anemometer traverse is usually used, with such a *correction factor* as may be necessary. When the air is being discharged from a grille or opening located at the end of a duct of the same nominal size as the grille, the

jet contraction at the opening is much less than in the case of a plenum outlet, and hence the anemometer factors are different also.

The 1939 edition of the Society's GUIDE [2] quotes the correction factors obtained by Davies [3] as follows: "for average use, with supply grilles, F can be taken as 0.97 at velocities from 150 to 600 fpm, and as 1.00 at higher velocities." This applies to a uniform traverse, using 6-in. squares, and with the anemometer held against the grille face. For calculating volumes the *effective* area is to be used, and this is defined as the average between the core area and the free area. (Using the definitions given on page 565 of THE GUIDE, 1939.) No mention is made of the type of approach. Davies' work was based primarily on small-duct approaches, *i.e.*, an approach duct the same size as the grille.

SUMMARY OF TESTS

Table 1 gives the results of over 50 anemometer tests on outlets of various kinds, in which the air was also measured by a Pitot tube or a calibrated orifice. These results show that the anemometer factor for a supply outlet may under certain conditions differ greatly from the values given in THE GUIDE, 1939. These tests indicated that the type of approach had a much greater effect on the coefficient than did the velocity, and that an outlet with a grille in place might behave quite differently from an outlet with no grille. Moreover the use of the *effective area* seems to be a complication of doubtful value.

Traverses of supply outlets may be made with other instruments than the rotating-vane anemometer but they are less convenient. The Velometer with a small *jet* is well suited to measuring the discharge velocity of nozzle-type grilles, because the velocity across the face of each nozzle is practically uniform. For a plenum supply outlet one other method is available, namely the treating of the entire opening as a metering orifice. Table 2 gives the orifice coefficients obtained by applying the common orifice equation to several types of outlets. Such coefficients may be used for calculating the capacity of an outlet when a given supply pressure is available in the plenum, or for obtaining the grille resistance when a given capacity is to be discharged. Manufacturers' resistance tables which deviate greatly from these values should be questioned. The orifice coefficient method is not applicable to openings located at the end of a small duct.

MEASUREMENTS AT INTAKE OPENINGS

Measurements of air velocity and quantity at the face of an air intake are in some respects more satisfactory than measurements at a discharge outlet. The velocity pattern at an intake is much more uniform, and there are no large variations due to upstream approach conditions. Hence such intake measurements are valuable in connection with tests of fans, unit heaters and coolers, warm-air furnaces and small air conditioners.

[2] ASHVE HEATING, VENTILATING, AIR CONDITIONING GUIDE, 1939, p. 828.
[3] Measurement of the Flow of Air Through Registers and Grilles, by L. E. Davies. (ASHVE TRANSACTIONS, Vol. 36, 1930, p. 201; Vol. 37, 1931, p. 619; Vol. 39, 1933, p. 373).

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TABLE 2—DISCHARGE COEFFICIENTS FOR AIR OUTLETS AND SUPPLY GRILLES MOUNTED ON A PLENUM OR LARGE DUCT

TYPE OF GRILLE	APPARATUS USED	VELOCITY RANGE, FPM	NOMINAL SIZE, IN.	CORE SIZE, IN.	PER CENT FREE AREA	NUMBER OF TESTS	DISCHARGE COEFFICIENT, C[a]
Free-open, no grille	*E, G*	400–1400	12 x 6	12 x 6	100	5	0.59
Punched sheet grille, oval openings	*D, E*	200–800	12 x 6	11 x 5	49	8	0.68
Punched sheet grille, square openings	*E, G*	700–1400	12 x 6	11 x 5	51	19	0.73
Assembled strip grille, fine mesh	*E*	400–1400	12 x 6	11 x 5	62	10	0.77
Bar grille, H and V bars on 1 in. centers	*E*	400–1400	12 x 6	11 x 5	66	6	0.77
Bar grille, H and V bars on ¼ in. centers	*G*	500–1700	24 x 8	....	72	10	0.76
Steel sheet with multiple round-entrance nozzles ⅝ in. diameter on 1 in. centers	*D, E, G*	500–2100	....	12 x 6	30.7	5	0.91
		500–2200	....	18 x 6	30.7	5	0.88
		500–2400	....	24 x 8	30.7	8	0.91
		500–2300	....	12 x 12	30.7	7	0.89

[a] C is defined by the equation: cfm $= 1096AC\sqrt{h/d}$, *where* h is the upstream total pressure in inches of water, d is the air density, and A is the free area or *daylight* area.

Note: Logarithmic graphs of these and other grille tests show an almost exact conformity to the square-root law.

TABLE 3—ANEMOMETER TRAVERSE DATA ON RECTANGULAR INTAKE OPENINGS
(*Free openings, no grilles*)

TYPE OF INTAKE	SIZE, IN.	FACE VELOCITY, FPM	APPARATUS USED	NUMBER OF TESTS	ANEMOMETER FACTOR, F[a]
End of duct (no flange)	14 x 5	500–1700	*G*	5	0.73
End of duct (no flange)	14 x 8	500–1500	*G*	5	0.73
End of duct (no flange)	14 x 10	500–1800	*G*	6	0.77
End of duct (no flange)	24 x 24	400–500	*G*	2	0.80
End of duct (no flange)	30 x 42	615	*H*	2	0.89
End of duct (no flange)	36 x 50	580	*H*	6	0.89
Square-edged wall opening	14 x 8	500–1500	*G*	7	0.83
Square-edged wall opening	24 x 24	400	*G*	3	0.92
Square-edged wall opening	30 x 42	515	*H*	2	1.03
Square-edged wall opening	36 x 50	575	*H*	4	1.03
Round-entrance wall opening	30 x 42	530	*H*	2	1.01
Propeller fan casings (square duct)	32 x 32 to 45 x 45	500–1200	....	18	Avg. 0.90

[a] F is the anemometer correction factor, defined by the equation: Actual cfm $= FAV$, *where* A is the total area traversed and V is the average velocity by anemometer reading.

For obtaining average velocities which may be used for volume calculations, the rotating-vane anemometer is again the most satisfactory instrument. Davies [4] reported correction factors of 0.762 to 0.832 when using the anemometer against the face of intake grilles of various designs and traversing to obtain an arithmetical average. Greene and Dean [5] derived a correction factor of 0.873 for typical exhaust grilles, when the traverse was made by a special averaging method according to Simpson's rule, giving greater weight to the center readings. The unusual nature of the method proposed by Greene and Dean, in addition to the fact that it is applicable to only a certain range of grille sizes, will limit its use. There is some question whether the anemometer calibration methods were comparable in the two investigations just cited. The 1939 GUIDE used Greene and Dean's coefficient of 0.873 *without* adopting their special traverse method.

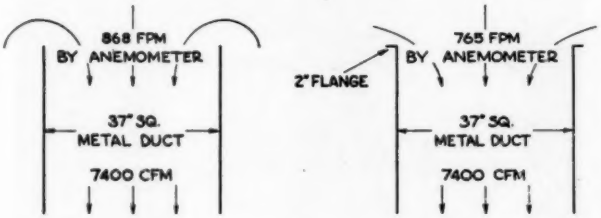

FIG. 12. EFFECT OF EDGE CONDITIONS ON INTAKE ANEMOMETER TRAVERSE

VARIABLES NEED STUDY

It seems that in studies of anemometer measurements at air intakes, two variables of great importance have been given very little consideration. One of these variables is the size and shape of the opening, the other, the edge condition or approach to the opening. Fig. 12 gives the results of an actual test showing the importance of small changes in the edge or approach conditions, even when the openings are large.

The tests reported in Table 3 show that a traverse of a large projecting intake duct does give the low anemometer factors obtained by Davies and others, but if the duct has a flange or if the opening is in a flat wall or ceiling, the coefficient or correction factor is close to unity. As the size of opening is reduced, the factor becomes smaller. The factor varies slightly with velocity as well as with the size, shape and edge condition of the opening.

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[4] Loc. Cit. See Note 3.
[5] The Flow of Air Through Exhaust Grilles, by A. M. Greene, Jr., and M. H. Dean. (ASHVE TRANSACTIONS, Vol. 44, 1938, p. 387.)

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Still another variable is the method of traverse. All data given in Table 3 and Fig. 12 were taken with a 4-in anemometer (though both 3-in. and 6-in. anemometers were used for check). Areas were divided into rectangles 4 in. to 5 in. on each side, and all traverses were simple cumulative readings, using 5 seconds to 30 seconds in each rectangle, depending on over-all size.

FIG. 13. METHOD OF TRAVERSING FREE AIR STREAMS

It is evident that the GUIDE recommendations for the use of anemometers give only approximate results as they stand at present, and that additional studies should be made to furnish accurate factors for all common conditions.

AIR-STREAM MEASUREMENTS IN OPEN ROOMS

Proper solution of the problem of air distribution in a room requires a knowledge of the spread and the throw of the entering air stream, and of the velocity distribution within it.

The included angle of spread of a straight-flow stream usually falls within the range of 12 deg to 25 deg for jets of air, steam and other fluids under a wide variety of conditions. Of course almost any desired angle larger than this may be produced by directional guide vanes, although the actual resulting stream angle is less than the angle of the vanes, and for uniform distribution the vane angles must be graduated across the face of the opening. Van Alsburg,[6] Mackey,[7] Jordan,[8] Theiss,[9] and others have reported values of the angle of spread from a straight-flow grille, and from such values a scale layout may readily be made for any room.

The throw of the stream depends on the angle of spread and on the initial velocity and quantity of air. The ceiling or the floor will interfere with the throw of warm or cold air streams. Maximum throw of the stream is usually defined as the normal distance from the face of the outlet to the plane in which the maximum velocity does not exceed 50 fpm. Experimental studies have shown that:

1. Maximum throw is directly proportional to average velocity at the supply outlet or grille.

2. Maximum throw is proportional to the square root of the free area of the supply outlet or grille.

These laws have been stated by Greenlaw and Hart[10] and others, and tests in the laboratories of Case School of Applied Science[11] indicate that they may be applied to most types of air conditioning grilles. Several other laws may be derived from those already given, and each of these laws aids in checking the accuracy of instrument applications and test methods.

AIR STREAM ANALYSIS

Another important principle applying to free air streams is that of conservation of momentum, which states that the product of the mass of the air moving times its average velocity across a normal plane, must be constant. This principle has been frequently used in such jet investigations as those of gas burners,[12] locomotive front ends,[13] and jet pumps. The principle is the same as the law of conservation of energy. From the conservation law both the throw of the stream, and the amount of room-air entrainment by it, may

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[6] Characteristics of Registers and Grilles, by J. H. Van Alsburg. (ASHVE TRANSACTIONS, Vol. 41, 1935, p. 245.)

[7] The Rationale of Air Distribution and Grille Performance, by C. O. Mackey. (*Refrigerating Engineering*, June, 1938, p. 417.)

[8] Air Diffusion in Ventilation Systems, by R. C. Jordan. (ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*, May, 1935, p. 253.)

[9] An Experimental Study of Air Entrainment in Room Ventilation, master's thesis, by Theiss. Case School of Applied Science.

[10] Air Paths from Grilles, by A. L. Greenlaw, and T. S. Hart. (*Heating, Piping and Air Conditioning*, June 1938, p. 387, July 1938, p. 452.)

[11] Measuring Air Distribution and Grille Performance in Air Conditioning, by G. L. Tuve. (ASHVE JOURNAL SECTION, *Heating, Piping and Air Conditioning*, November 1937, p. 700.)

[12] Bureau of Standards Technologic Paper No. 193, Design of Atmospheric Gas Burners.

[13] A Study of the Locomotive Front End, by Young. (University of Illinois, Engineering Experiment Station Bulletin No. 256.)

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be calculated. Mackey [14] has calculated that the air entrainment is 11 to 1 at the end-plane of the throw. This is based on assumptions of uniform velocity across the stream, 14 deg spread, and 600 fpm initial velocity across an area of one square foot. But the assumption of uniform velocity is in error, and a more accurate experimental verification of the conservation law is possible by using an indicating-ribbon grid and a Velometer, as shown in Fig. 13. Table 4 gives results of such an analysis by Theiss,[15] for two of the seven

TABLE 4—RESULTS OF AIR-STREAM ANALYSIS BY THE CONSERVATION LAW

TYPE OF OUTLET	FREE OPENING IN PLENUM	OPEN END DUCT
Size of outlet, inches	14 x 8	14 x 8
Rate of flow, cubic feet per minute	1161	1156
Horizontal spread, total angle, degrees	23.5	24.5
Measurements in a normal plane 3 ft from face (see Fig. 13):		
Mean velocity feet per minute	367	316
Cross-sectional area of stream, square feet	7.3	7.8
Air entrainment, per cent of primary air	478	347
Summation of momenta in 6 in. squares	176,600	117,600
Measurements in a normal plane 6 ft from face:		
Mean velocity, feet per minute	327	250
Area of stream, square feet	13.3	14.0
Entrainment, per cent of primary air	567	472
Summation of momenta in 6 in. squares	181,700	118,900
Measurements in a normal plane 12 ft from face:		
Mean velocity, feet per minute	238	202
Area of stream, square feet	28.2	27.0
Entrainment, per cent of primary air	800	588
Summation of momenta in 6 in. squares	178,700	115,400

straight-flow outlets which he investigated. These results, involving thousands of observations and calculations, demonstrate the reliability of the method, and of the Velometer, for air-stream measurements in rooms. The difference in air entrainment in the two cases of Table 4 is noteworthy.

In considering the use of other instruments than the bridle-vane anemometer (Velometer) for measuring air velocities in open rooms, the following suggestions may be helpful:

1. Where directional effects are important, use a mechanical-type instrument.
2. Where comfort effects are important, and ample time is available, use a heated-type instrument.
3. For relative measurements, adhere to a single instrument, selecting one that will cover the entire velocity range.
4. For absolute measurements (*i.e.*, those in which an accurate result in feet per minute is required), check the measurements by using more than one type of instrument.
5. Always take the average of several readings to represent the final result.

[14] Loc. Cit. See Note 7.
[15] Loc Cit. See Note 9.

DISCUSSION

PROF. D. W. NELSON (WRITTEN): The authors have presented their material in good shape and at a very opportune time. Where special inclined manometers are mentioned, would it not be well to describe what constituted their peculiarities, as to gage fluid, slope, bore of tube, divisions and extent of scale? If they are commercially available this might not be necessary. Also would it not be well to mention or describe the type of micro-manometer used in calibration?

The statement is made that the heated thermocouple gave a definite concave upward calibration curve. In Fig. 10 the curvature is based on only one point, the only one shown at higher velocities.

The paper gives a very poor impression of the Kata-thermometer due largely to selection from a careless manufacturer. The six or eight available to us have all been reasonably correct and very consistent.

The subject of air stream behavior with only slight reference to air meters has been considered, and one statement reads, "Several other laws may be derived from those already given, and each of these laws aids in checking the accuracy of instrument applications and test methods." The basis for these laws is in a paper [16] which was published in *Heating, Piping and Air Conditioning* and not presented before the Society, and in a thesis at Case School of Applied Science. It is to be regretted that the data to support these statements of laws have not been presented in a paper on air streams before the ASHVE.

PROF. A. E. HERSHEY (WRITTEN): The material contained in this paper is very timely and important. The statements which I believe need some amplification or clarification are those regarding air flow in jets. I am inclined to agree with Professor Nelson that this material is somewhat out of place in this paper and that it should be presented in a more complete form in another paper, dealing exclusively with flow in jets. In any case the statement that the principle of conservation of momentum is the "same as the law of conservation of energy" should be changed to "similar to . . ." or "Analogous to the law of conservation of energy." It also might be desirable to modify the next statement to indicate that in addition to the principle of conservation of momentum, it is necessary to make some assumption regarding the angular spread of a jet before it is possible to calculate the entrainment and throw of a stream or jet, since conservation of momentum alone does not yield these results.

We should appreciate having more detailed information regarding the inclined manometers and the micro-manometers included in the paper, but this information may not be of sufficient general interest to justify its inclusion.

PROF. M. K. FAHNESTOCK (WRITTEN): This paper contains a large amount of information which should prove to be useful to a large number of people and we believe that the authors should be complimented on assembling and presenting the data so that they are available to those in the air conditioning industry.

Would it not be advisable to include a fourth purpose of air-velocity measurement to include the determination of air volumes or air quantities? Also could the authors insert the numerical value of the minimum velocity for which the statement of the 1 per cent accuracy holds? Were straighteners used between the fan and the Pitot tube traversing stations in the metering tunnels? If so, perhaps they should be shown on the sketches.

The difficulty of obtaining and maintaining still air in a large space might also be included as one of the complications making the use of the moving instrument calibration method impractical.

[16] Loc. Cit. Note 10.

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Referring to Fig. 8 and certain places in the text, if possible, I believe it would be desirable to standardize on the notation for designating the rotating vane or propeller type of anemometer, and also on the deflecting-vane or bridled-vane anemometer.

In Table 1 I would like to know if A is the total area traversed as in Table 3? Should it not be designated in the table and in the text that the anemometer used was a rotating vane or propeller type? I believe it would be desirable to give more detailed information as to the locating of the anemometers relative to the openings and the grilles, both supply and exhaust, the number of readings or positions in a traverse, and the time allowed for each reading. One more question, is the supply pressure the static pressure in the plenum?

C. H. RANDOLPH (WRITTEN): Was the Kata-thermometer used a high or low temperature Kata? I happen to have one of each and am interested in knowing what your opinion is of one compared to the other.

J. L. BLACKSHAW: There are three questions of interest to me, namely: What type of handle did the anemometer have; its position on discharge test; what was static pressure, and how was it taken?

AUTHORS' CLOSURE: In answer to the questions by Professors Nelson and Hershey, the inclined manometers used in the low-pressure tests were commercial instruments reading 0.50 in. of water over a scale length of 10 in. They were filled with pure alcohol, and were calibrated by comparison with a micrometer-manometer having an inclined leg with an optical magnifying system. This latter standard gage could be read accurately to 0.0005 in. of water.

The two low-temperature Kata-thermometers were not very satisfactory, and a third one furnished by the same manufacturer did not agree with either of the other two. The performance of these instruments was reasonably consistent but the calibration factors furnished by the manufacturer were evidently incorrect.

We are glad that Professor Fahnestock mentioned the item of straighteners, because these were inadvertently omitted from the sketches. All the test units in which Pitot tube traverses are indicated were equipped with straightening vanes made and located in accordance with the specifications in the ASHVE Fan Test Code.

In answer to Mr. Blackshaw, the anemometers were always held at the end of a long streamlined handle about ½-in. in diameter, and the observer was careful to stay out of the air stream.

While this paper dealt primarily with instrument calibrations, Tables 1 to 4 are in the nature of a progress report or preliminary data on instrument *applications*. These results will be supplemented and correlated with further data to be presented in later reports.

No. 1141

A STUDY OF CERTAIN FACTORS THAT AFFECT THE DROOP CHARACTERISTICS OF THERMOSTATS

By BURTON E. SHAW * (*MEMBER*) AND LAVERNE LYON,** GOSHEN, IND.

IT is the purpose of this paper to study some of the factors affecting the droop characteristics of thermostats. Droop may be defined as the peculiarity of a thermostat to control the temperature adjacent to its location at a point below the setting (dial adjusment) under certain conditions.

Droop may be affected by artificial heat developed within the thermostat, wall-temperature conditions, thermostat mass, air drift through the thermostat, heat-gain rate of the system, heat-loss rate of the system, and many other factors. Because of the necessary limitations of the paper the authors will confine their comments to a study of droop conditions occasioned by using artificial heat within the thermostat, the effect of cold wall conditions, and the effect of variations in the rate of indoor temperature changes.

The rate of heating and cooling in an average residence may vary from one degree Fahrenheit up per hour to 10 F or more, depending on weather conditions, insulation, and other factors. In order to study the droop for various rise and fall rates special test equipment was constructed as shown in Figs. 1, 2, 3 and 4.

Essentially the test apparatus consisted of a large cooling room within which a smaller insulated structure housed the thermostat, heating means and air velocity regulator. The temperature in the cooling room was maintained by a compressor with an adjustable thermostatic control. The heating means in the smaller enclosure which simulated the residence to be tested was controlled through a relay by the thermostat located therein. The heat input was regulated by a variable voltage transformer located externally to both enclosures. Temperatures within the residence were measured on a recording instrument with a 60—80 F scale calibrated to 0.2 F. A temperature recorder in the larger enclosure (outdoors) was also used.

The small structure was lightly insulated so that the rates of temperature changes inside could be regulated with a minimum change of outdoor tempera-

* Research Chief, Penn Electric Switch Co.
** Research Engineer, Penn Electric Switch Co.

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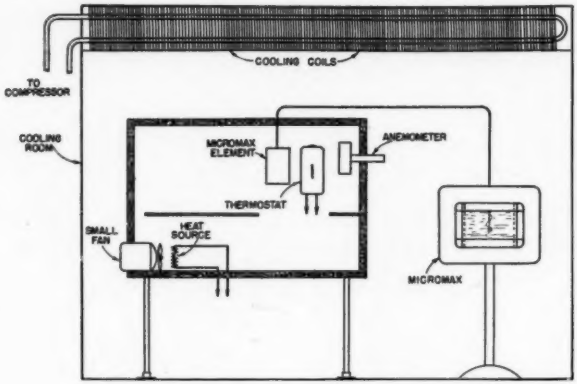

FIG. 1. TEMPERATURE DROOP TEST APPARATUS

ture. Thus the radiation effect to cold outside walls and windows was minimized. Cold wall mass effect was eliminated by hanging the thermostat in air near the center of the box.

The capacity of the thermostatically controlled cooling system was large enough to hold a constant outside temperature despite any influence of the heat loss from the smaller structure.

In the field a variable amount of under-run and over-run of the heating system may occur. With the apparatus used this factor was reduced to a minimum thus eliminating another variable.

Cold walls were made the subject of a separate test. The method and apparatus used is shown in Figs. 5, 6 and 7. The apparatus consisted of an

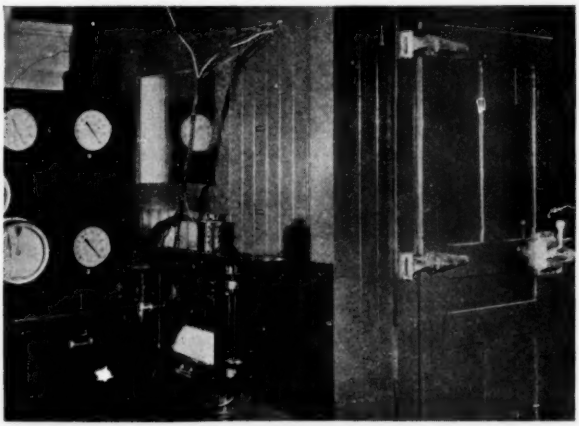

FIG. 2. EXTERIOR VIEW OF COOLING ROOM

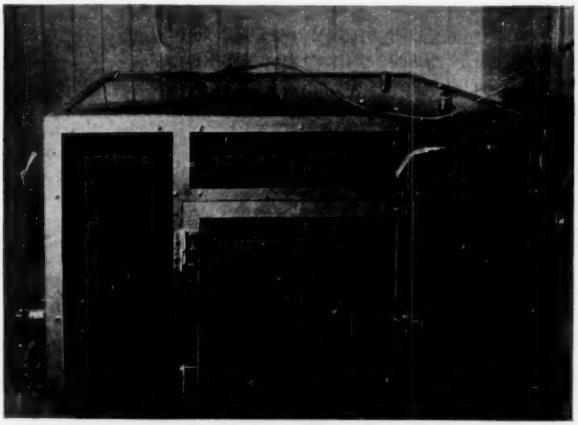

FIG. 3. VIEW OF INSULATED STRUCTURE HOUSING THERMOSTAT

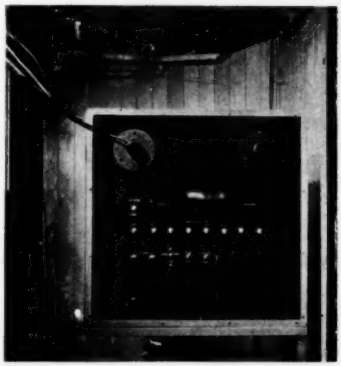

FIG. 4. END VIEW OF CONTROL PANEL ON INSULATED STRUCTURE

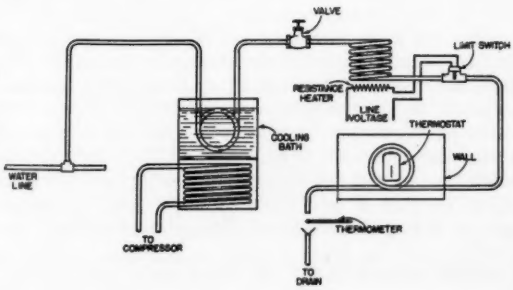

FIG. 5. COLD WALL TEMPERATURE TEST APPARATUS

insulated wall on which the thermostat was mounted. Within the wall was located a water coil through which water at a constant temperature was circulated. The temperature of the water was regulated by means of a limit switch controlling within 0.1 F. The thermostat controlled the ambient temperatures adjacent to itself and the wall.

Throughout the tests outlined artificially heated thermostats were used. The thermostats could be adjusted readily to give differentials ranging from conventional control (3 F or more) to line temperature control [1] (½ F or less)

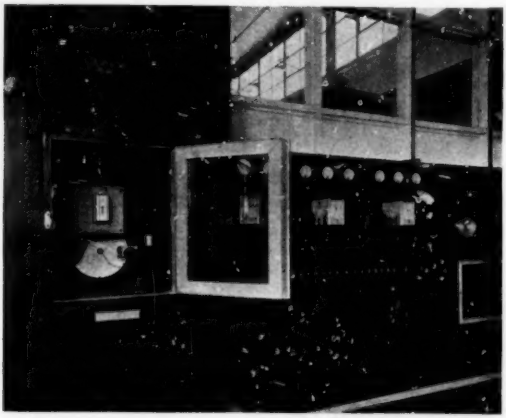

FIG. 6. VIEW OF COLD WALL TEMPERATURE TEST EQUIPMENT

Various designs are resorted to by different manufacturers but suffice it to say that in all cases investigated the results obtained did not differ materially from those reported on except in cases wherein the thermostat was definitely defective in design or performance.

RESULTS

The rate of temperature rise in degrees per hour plotted against the operating temperatures of the thermostat is shown in Fig. 8. By operating temperature is meant the maximum and minimum temperature of the controlled air adjacent to the controlling thermostat. In other words, the ambient temperatures at which the thermostat opens and closes. To obtain a variation in temperature rise the heat source in the small structure corresponding to the residence was varied to produce average rates of rise of indoor temperature from 1 to 16 F per hour. The outdoor temperature was held constant to give a 6 F per hour fall. Thus, the sole effect of variation of temperature

[1] Comparative Study of Combustion Results with Various Thermostats, by B. E. Shaw. (ASHVE TRANSACTIONS, Vol. 42, 1936, p. 60.)

rise on the operation of the thermostat is shown by these curves. Two tests were run, one without an artificial heater, and the other with an artificial heater. The thermostat operating without an artificial heater corresponded to a conventional type thermostat except that its differential was somewhat wider than will ordinarily be found with these thermostats.

Identical tendencies of the operating characteristics occur with various differential settings of a conventional thermostat. For the purpose of obtaining the total influence of the artificial heat on the thermostats tested the mechanical

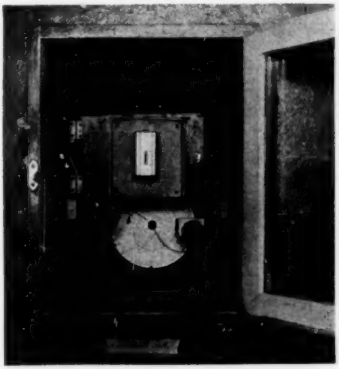

FIG. 7. CLOSE-UP VIEW OF COLD WALL APPARATUS

differential was held the same for the tests with and without the artificial heater. This resulted in greater contact pressures and snap of contacts as compared to a conventional unit.

The heaters consisted of a resistance wire series connected with the load, whenever the thermostat called for heat. The resistance wire supplied approximately one-fourth watt of artificial heat to the thermostat. For the test without artificial heat a copper jumper was substituted for the resistance wire.

An analysis of these curves follows: Curves *A* and *C* show the opening and closing temperatures with artificial heat. With the slower rates of heat rise and consequent longer *on* periods, the closing temperatures of the thermostat decreased slightly more than 1 F below the true setting (closing temperature in water) at a temperature change approaching zero degrees per hour. This means that artificial heat is retained in the thermostat throughout the *off* period, and a greater lowering of temperature outside the thermostat is necessary to close it. Curves *B* and *D* show the closing and opening temperatures when the artificial heater is removed. The difference between Curves *A* and *B* is a measure of the heat retention in the mass of the thermostat due to the artificial heater. Curves *C* and *D* show considerable drop of the opening temperatures of the thermostats as the rate of temperature rise decreases. Curve *C* drops somewhat more acutely than does Curve *D*. The difference is due to

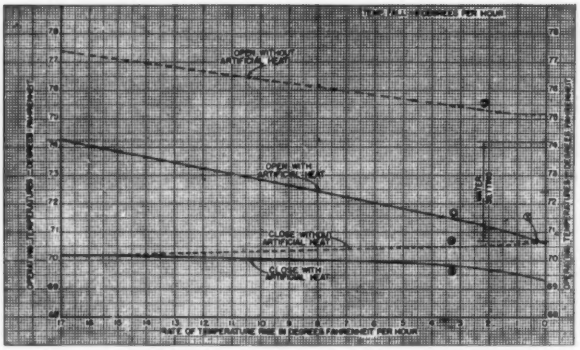

FIG. 8. EFFECT OF RATE OF TEMPERATURE RISE ON A THERMOSTAT

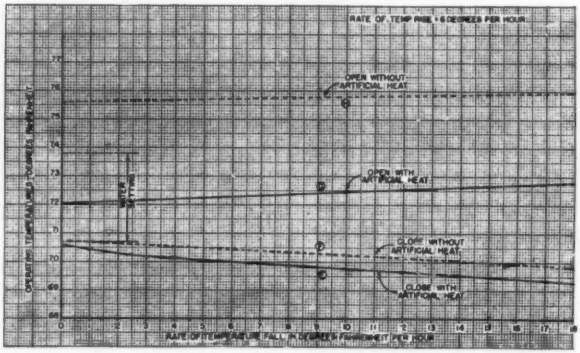

FIG. 9. EFFECT OF THE RATE OF TEMPERATURE FALL ON A THERMOSTAT

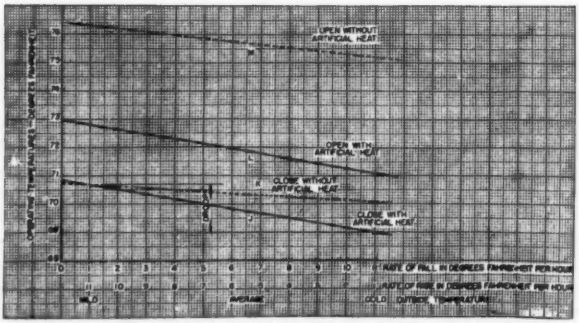

FIG. 10. TEMPERATURE DROOP OF A THERMOSTAT COMPUTED FROM CURVE

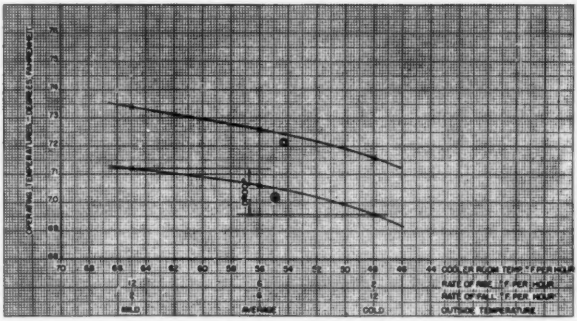

FIG. 11. TEMPERATURE DROOP OF A THERMOSTAT ACTUAL TEST

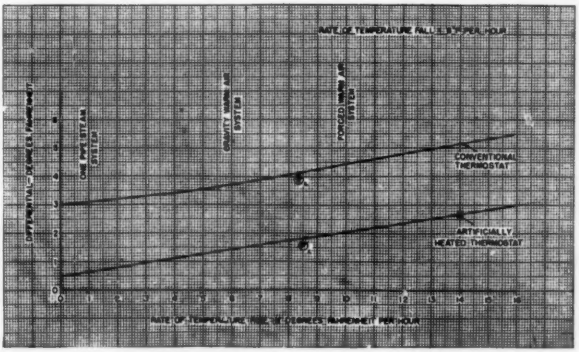

FIG. 12. DIFFERENTIAL OF A THERMOSTAT WITH VARIOUS RATES OF RISE

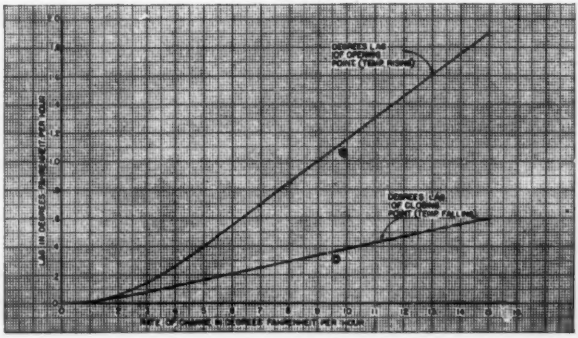

FIG. 13. LAG OF THE THERMOSTAT BEHIND THE AMBIENT TEMPERATURE, NO ARTIFICIAL HEAT

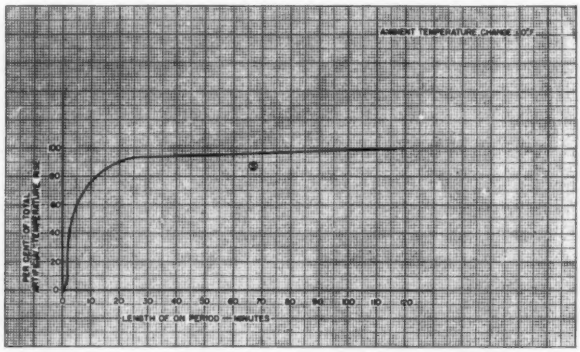

FIG. 14. TEMPERATURE RISE WITHIN THE THERMOSTAT FROM THE ARTIFICIAL HEATER

the artificial heater used in obtaining Curve *C*. The decrease of both Curves *C* and *D* was the result of the natural lag of the thermostat behind the rise in temperature. Decrease of Curve *C* more than Curve *D* shows that during a short *on* period the artificial heater did not increase the artificial temperature

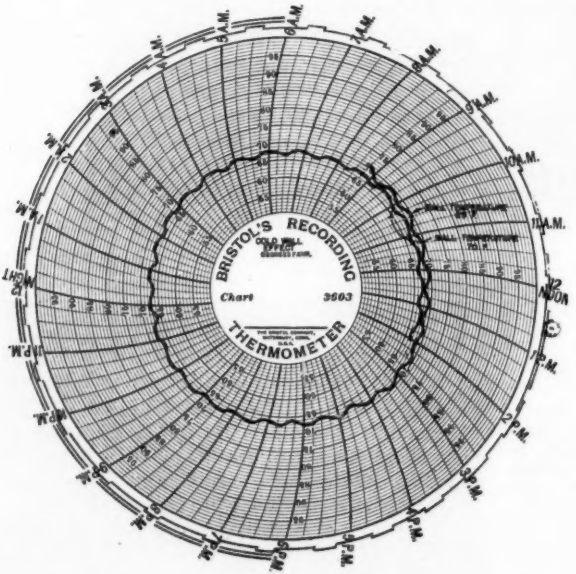

FIG. 15. ROOM TEMPERATURE CHART SHOWING COLD WALL EFFECT

within the thermostat as much as it did during longer *on* periods. Point *X*, the intersection of Curve *C* with the closing point of the thermostat in water shows that an *on* period of four or five hours was necessary before the thermostat opened below its true setting. Curve *D* straightens out near the ordinate axis at low rates of rise indicating that up to about a 2 deg per hour change the thermostat was able to follow air temperature changes very closely. Fig. 9 shows the operating characteristics of a thermostat as altered by the variations in indoor temperature fall only. These data were then added to Fig. 8 to obtain Fig. 10.

Fig. 10 is a mathematical combination of Figs. 8 and 9 giving the normal droop of the thermostat from mild outside temperatures to cold outside temperatures. Under normal conditions the rates of rise and fall of inside temperature vary conversely as the outside temperature changes. Therefore, by adding the Curve of Fig. 8 to Fig. 9 within the limits of 1 to 11 F per hour rate of temperature change, an effect equivalent to a range of outdoor temperatures is produced on the thermostat. This particular rate range was chosen arbitrarily. Any greater or smaller range might have been used resulting in more or less droop of the thermostat. In the field a different range of temperature change is found with every individual installation. However, since the slower changes had the most effect on droop it was felt that the values used in this paper produced close to the maximum.

As shown in Fig. 11 the temperature droop in one continuous test where the outdoor temperature was varied gave practically identical results as compared to Fig. 10. This droop amounted to approximately 3 F maximum for the total range investigated. It is to be noted from Curve in Fig. 11 that the changes in rates from mild to cold weather were produced by total outside temperature change of from 65 to 48 F. The same change in rates in an actual field installation may be the result of an outside temperature change of from 65 F to below zero depending on the heating system, building, insulation and other like factors.

Fig. 12 was adopted from Fig. 8 and shows the changes in differential under various rates of temperature rise. Different heating systems produce a rate of temperature rise somewhat in the order shown in Fig. 12. Thus, the same thermostat on one setting or adjustment may have a differential of from ½ to 3 F (if it is of the artificially heated type) depending on the type of system used. It is not uncommon for a field installation to require an adjustment of the thermostat differential to suit the particular requirements of that installation. The velocity of air motion produced by various systems has some effect on the differential of the thermostat and also on droop. A study of this factor is being reserved for a later paper. These tests were made at an air velocity of approximately 8 fpm.

Fig. 13 shows temperature lag of the thermostat as a whole behind the ambient air temperature at various rates of change when no artificial heat was used. The opening and closing temperatures lagged because of a combination of the slow infiltration of the ambient air through the thermostat cover and the thermal lag of the bi-metal.

Fig. 14 is based on the effect of artificial heat on the opening point of the thermostat, and is taken from Fig. 10. It shows the temperature rise within

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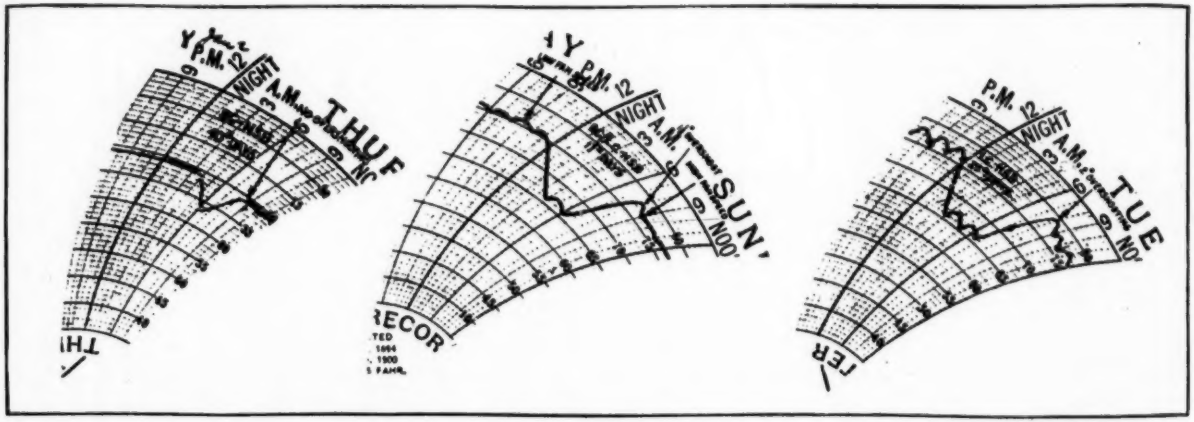

FIG. 16A. ROOM TEMPERATURE CHART SHOWING NIGHT SETBACK WITH ARTIFICIAL HEAT

FIG. 16B. ROOM TEMPERATURE CHART SHOWING NIGHT SETBACK WITH OVERSHOOTING

FIG. 16C. ROOM TEMPERATURE CHART SHOWING NIGHT SETBACK WITH OVERSHOOTING

the thermostat resulting from the artificial heating. The difference between the opening curve with and without artificial heat is caused by the temperature rise in the thermostat resulting from heat artificially put in. The important point to be gained from this curve is that most of the effect of the artificial heat is felt within the first 20 min of operation. For longer *on* periods the curve straightens out with only a very slight rise. An ideal heater would have all of its effect felt immediately after the thermostat closes and thereafter supply a heat input of just enough to hold the temperature gained, thus reducing the possibility of the thermostat opening without a rise in room temperature. Fig. 14 shows that the thermostat tested approached this ideal very closely.

COLD WALL EFFECT

Fig. 15 is a room temperature chart showing the changes that occurred in the operating temperature of a thermostat when the temperature of the wall on which the thermostat was mounted varied. A change of 5 deg in wall temperature caused a 2 F change in the operating points of the thermostat, both closing and opening. As the temperature of the wall went up from 65 to 70 F the air temperature as controlled by the thermostat decreased. Since the wall temperatures ordinarily decrease as the outside temperatures become colder, the thermostat will operate at a high temperature. Thus, cold wall effect is opposite to the effect of the rate of indoor temperature change. Droop resulting from one is at least partly off-set by the other.

Fig. 16 shows sections of room temperature charts where the operation of night-setback (NSB) under various conditions took place. These were taken from the field during previous investigations by the authors. Fig. 16A depicts night-setback with sufficient artificial heat in proportion to the normal differential of the thermostat to produce nearly straight-line temperature operations. No over-run occurred because of the snubbing effect of the artificial heat helping the thermostat to open at its normal setting despite the fact that cold walls and thermal inertia of the thermostat tended to keep it closed and cause over-shooting.

In Fig. 16B and C over-shooting occurred where the ratio of the artificial temperature rise to the real differential of the thermostat was smaller allowing wider actual differentials. In other words because of the lag in bi-metal temperature from cold wall conduction and slow infiltration of ambient air the room temperature was allowed to rise above the temperature setting of the thermostat before it opened to shut down the heating system.

CONCLUSIONS

1. From the standpoint of thermostat design, droop is caused primarily by slow infiltration or drift of air into the thermostat and thermal inertia of the bi-metal and thermostat mass. This factor was equally effective in both artificially heated and conventional designs.
2. Some droop is caused by the use of artificial heat in a thermostat because of heat retention or carry-over through the *off* period after moderately long *on* periods. Another factor of least importance in producing droop is the effect of artificial heat on the opening temperature of the thermostat. With a correctly designed heater

most of the artificial temperature rise within a thermostat takes place during the first few minutes of the *on* period. Thereafter the temperature gain is slight and has little further influence on the droop of the thermostat.

3. The effect of cold walls on droop is opposite to that of the factors of thermostat lag and artificial heat. As the weather becomes colder and the wall temperatures decrease the thermostat influenced by such cold walls tends to operate at higher temperatures. Thus the effect of cold walls is to partly off-set the droop of a thermostat.

4. Artificial heat in thermostats tends to decrease the morning over-run after a setback of temperature at night caused by cold walls, and air drift lag of the thermostat.

5. Test data not yet completed indicate all types of artificially heated thermostats regardless of design or circuit method apparently will function to give the same general characteristics. With one type, however, the results were negative due to improper design and faulty performance.

No. 1142

THE APPLICATION OF STORAGE REFRIGERATION TO AIR CONDITIONING

By CARL F. BOESTER* (*MEMBER*), ST. LOUIS, MO.

THE idea of accumulating or storing up refrigeration over a given period of time for later use is not new. As a matter of fact, the idea has been experimented with for so long that it is impossible to ascertain who should rightfully be given credit for it. In recent years, however, considerable research on design and development has been done in this line with a view to applying the principle to air conditioning, primarily for the purpose of lowering first costs, and secondly, for the purpose of lowering operating costs of air conditioning.

Several methods of applying storage refrigeration to air conditioning have been developed and reported by Leopold, Friend, Freund, Betz, Elmer, Marston and Pfeiffer. The research activities of these men have been along various lines, such as the storage of refrigeration in the form of chilled water, chilled brine and ice. Reports have been made in technical and trade publications. The author's experience has been along the line of accumulating refrigeration in the form of ice on evaporator coils.

Whether or not storage refrigeration is applicable to a particular air conditioning problem depends upon the load. To begin with, the load is determined in the usual manner for any type of installation, using the engineer's own method of heat gain calculation. After the size of the job in tons of refrigeration is determined, the next step is to determine whether or not storage refrigeration is desirable from an application engineering standpoint and, second, whether or not it is economical.

The average commercial or industrial air conditioning load operates only a few hours out of each 24-hour period, and during this time refrigeration is required at varying rates from zero to peak capacity. This brings in two factors—hours of operation and usage or load factor, giving an equation for determining the amount of refrigeration to be consumed, which may be expressed as follows: tonnage (as indicated by the heat gain survey), times hours of operation, times usage or load factor, divided by 24—the result being the

* Air Conditioning Engineer.

Presented at the Fall Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Atlanta, Ga., October, 1939.

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amount of refrigeration to be consumed as distinguished from the rate at which it is to be delivered.

Since one ton of refrigeration may be assumed as the equivalent of the production of one ton of ice in 24 hours in a storage refrigeration application of this type, the result of the equation is the equivalent of the total tons of ice that will be consumed and that will, therefore, have to be provided. The

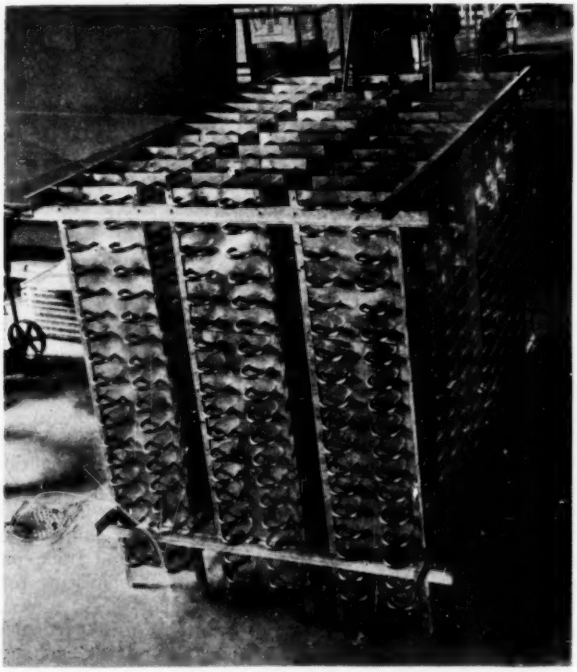

FIG. 1. TYPICAL 3-TON ICE ACCUMULATING EVAPORATOR FOR STORAGE REFRIGERATION PURPOSES

refrigerating equipment applying this storage principle must, then, be selected to store up this amount of refrigeration in the form of ice.

Before discussing the means of accumulating the required refrigeration in the form of ice, it is well to consider the factors affecting the equation. To begin with, the tonnage as shown by the heat gain survey expresses the maximum rate at which refrigeration is required for the air conditioning load, and any storage refrigeration system has to be able to deliver refrigeration at the maximum rate as well as at any percentage of this rate, according to particular load requirements. The hours of operation affect the amount of refrigeration to be consumed. The ratio of operating period to accumulating period determines the feasibility of storage refrigeration. Thus, if the accumulating period

is 24 hours and the place of business to be cooled requires cooling for eight hours each day, then there is a ratio of three to one. The guiding principle might be expressed as follows: the longer the hours during which cooling is required as compared to the accumulating or recovery period, the less desirable storage refrigeration becomes. If the period over which refrigeration can be accumulated is 72, or possibly 144 hours (as in storing refrigeration for church cooling), the applicability of storage is most apparent; also if the accumulating period is limited to the hours from 6 p.m. to 6 a.m. because of an attractive off-peak rate offered by the power company during this period, the principle may still be applicable.

The other governing factor in the applicability of storage refrigeration is that of usage, generally expressed as *percentage of peak*. For example, in theater air conditioning, since people comprise most of the load and since the number of people in the theater varies, the usage or load factor also varies as much as from 20 to 90 per cent of the peak or design load. Careful analysis of theater attendance records indicates that for a theater operating 12 hours a day, the usage factor is approximately 55 per cent. For theaters operating only during the evening, the usage factor is 65 to 70 per cent.

Thus, if the hours of operation are long but the usage factor is low (as is often the case in cafeterias and restaurants), or if the hours of operation are short but the usage factor is high (as in the case of church cooling, or, in the case of neighborhood theater operation), then storage refrigeration is quite desirable. When the usage factor is high and the hours of operation are long, it is not practical to design an air conditioning system employing storage refrigeration.

It might be well to mention that determining the usage factor sometimes presents difficulties, but, in addition to the expedient of making a personal survey, it can often be obtained from manufacturers of equipment for storage refrigeration applications.

Assuming, now, that the equation indicates that a given set of conditions makes it desirable to employ storage refrigeration, the engineer is presented with the problem of determining the best way to select and design equipment for the application. If, after analysis, it is considered desirable to store the refrigeration in the form of ice rather than in the form of brine or chilled water, a simple way to accumulate ice is upon evaporator coils submerged in a tank of water.

In considering the tank for this purpose, the question of insulation immediately arises, and whether or not insulation should be used depends upon an analysis of tank losses as compared with insulation costs. This must be determinded by the application engineer in arriving at his total cost. Often the tank is placed in the ground where the temperature the year round rarely exceeds 55 F. Thus the temperature difference between the ground and the ice is rather small, and this, multiplied by the square feet of tank surface will give the tank loss.

It has long been known that approximately 80 to 100 sq ft of evaporator surface times the coefficient of heat transfer is required to accumulate one ton of ice at an economical cost. It can be appreciated that the thicker the ice on the evaporator, the lower the refrigerant temperature in the evaporator must be in order to accomplish the desired rate of accumulation and counteract the effect of the accumulated ice acting as insulation. Research indicates that

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an accumulation to a thickness greater than 1½ in. to 1¾ in. on any type of evaporator surface is not economical.

In designing an evaporator with the required amount of surface, the following must be borne in mind: the amount of refrigerant charge, the refrigerant temperature, the utilization of all of the evaporator surface and the progressive use of such surface, the design of the evaporator to withstand expansion of the ice as it freezes, the weight of the ice on the evaporator section, and the displacement of the evaporator itself and of the evaporator with the required amount of ice accumulated on it in ratio to tank volume.

From the standpoint of lower first cost and lower operating cost, it has been found practical to design an evaporator approximately as outlined. It should be constructed of ⅝-in. tubing because the necessary prime surface in tubing of this size will displace the minimum amount of water in a tank, and thereby require a tank of smaller capacity. For example, 80 sq ft of surface will require approximately 500 lineal feet of ⅝-in. tubing. This might well be made up into an evaporator with tubing on 3½-in. centers having over-all dimensions of 14 in. wide, 63 in. high and 84 in. long. An evaporator of the same capacity made in 1¼-in. tubing would be some 20 in. wide, some 10 ft long and some 80 in. high, and would require a tank having twice the volume of the evaporator using smaller tubing. Such an evaporator almost doubles the tank cost.

The smaller size tubing has another advantage. A ton of ice has a displacement of about 37 cu ft. The more nearly an evaporator coil approaches this volume in accumulating a ton of ice, the more desirable it is, for as the ice is reduced to low water temperatures, the volume of water in the tank becomes disproportionate to the volume of ice in the tank. When this occurs, it becomes extremely difficult to maintain and control low water temperatures.

The method of feeding the refrigerant is of great importance from the standpoint of minimum refrigerant charge. The greatest efficiency has been obtained by the use of multiport expansion valves set at three or four degree superheat. This low superheat setting does not permit the use of any of the evaporator section as a drier; therefore, a heat exchanger is required to accomplish this function. The refrigerant charge and method of feed should be such that the ice is accumulated progressively on the evaporator section, and under such progressive accumulation, it is possible to superheat the refrigerant vapor in portions of the evaporator on which there is no ice accumulation, thereby maintaining higher and more efficient suction temperatures conducive to low operating cost.

An evaporator made of light ⅝-in. tubing is able to flex with the expansion of the ice as it freezes without damage to the evaporator, and the evaporator, with its accumulation of ice upon it, has a tendency to float in the tank of water due to the fact that the ice is lighter than water. Therefore, the evaporator itself does not have to bear the weight of the ice and is sufficiently strong even though of such light material.

The most important factor governing the successful application of storage refrigeration to air conditioning is the ability to melt the ice and to control its melting rate according to the load demand. It is at this point that experience in this connection is valuable. While anyone can accumulate ice with varying degrees of success, the ability to melt it at the required rate is all-important. It is quite possible to have a tank two-thirds full of ice and yet

withdraw water from the tank at a temperature of 60 to 65 F, entirely useless for air conditioning purposes. The volume of water circulated through the tank, the temperature range over which the water is to be cooled, the velocity of the water through the tank, the control of the water flow through the tank, and the number of square feet of ice surface exposed to the flow of the water, very definitely affect the ability to control the rate of melting, and thereby the success of the storage refrigeration application. Data pertaining to the rate of heat transfer (K) between the ice and the water under various water temperatures, water velocities and square feet of ice surface required to accomplish this result have, as a rule, been considered a trade secret of the manufacturers supplying this type of equipment. The real success of a storage system hinges on these factors. Too rapid melting and thereby too low a temperature wastes refrigeration, and too high a temperature, of course, will not produce the desired results. The manufacturer of ice accumulating evaporators usually breaks the evaporator up into a number of sections in order to provide the correct amount of surface for a given flow of water, and should be consulted on problems of this nature and relied upon for guarantees of performance under given load conditions.

As to operating costs, a ton of ice can be accumulated with approximately 22 kwhr of electrical energy. This is comparable to a ton of refrigeration delivered by other means. The storage refrigeration equipment functioning at high pressure operates at a constant load and as a rule takes advantage of lower condensing temperatures available at night, if a cooling tower or an evaporative condenser is used. On the other hand, a conventional direct-expansion job with frequent cycling and partial load inefficiencies uses the same amount of power. Thus, where no demand charge is made, the power costs of direct expansion and of storage refrigeration are comparable. However, where an off-peak rate is available, or a high demand charge exists, storage refrigeration reflects a considerable saving in operating cost.

As to first cost for the equipment, savings when employing the storage principle have been as high as 70 per cent, and they average about 25 to 30 per cent. On those jobs where hours of operation are long and the usage factor high, storage refrigeration is hardly feasible, and for such a job the first cost would be high.

Storage refrigeration permits many new application engineering practices which are conducive to low first costs. These are too numerous to mention but have become obvious when a comparison of costs is made for a given job. Each individual job must be analyzed to establish whether or not it can best be handled by the application of storage refrigeration or done with full capacity, direct expansion equipment.

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DISCUSSION

F. J. DEAN, JR.: How is the problem of oil accumulating in the evaporator handled?

CARL F. BOESTER: The only place for oil is in the crankcase of the compressors. Recently designed oil separators placed in the compressor discharge line can properly correct the oil problem. Evaporator volume and refrigerant velocities, in the evaporator of the author's design, make it practically impossible for oil to return unless a very large volume of oil is present. Such volume would foul the evaporator and render it inefficient. With the use of a good oil separator, no oil was found in an evaporator after months of use; furthermore, no reduction of oil level in the crankcase was noted.

MR. DEAN: What method of controlling ice accumulation is used?

MR. BOESTER: Control of ice accumulation is by the use of a remote bulb thermostat placed 1½ in. from the evaporator tubing. This becomes encased in ice and is set at 28 F. When that temperature is reached, the thermostat shuts a main liquid line magnet valve. The compressor starts and stops on pressure.

No. 1143

A STANDARD AIR FILTER TEST DUST

By FRANK B. ROWLEY * AND RICHARD C. JORDAN ** (*MEMBERS*),
MINNEAPOLIS, MINN.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the Graduate School of the University of Minnesota in cooperation with the Engineering Experiment Station, University of Minnesota.

THIS paper presents the results of research which supplement the work described previously in papers [1, 2] delivered before the Society. The object of this part of the program was to study the requirements of a standard test dust for filters to be tested on a weight basis, and to investigate the properties of different dusts in relation to those requirements. The test apparatus and procedure were described in the previous papers. The apparatus, with the exception of the dust feeding device, was the same as that described in the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work. The photograph, Fig. 1, shows a general view of the test equipment as set up and used throughout the tests.

The four filters selected in the previous test program as typical of those in practice were used throughout these tests. These filters have been designated by the letters *A, B-2, C,* and *D* and were described in detail in the first paper. Briefly they may be described as follows:

A—A permanent type of cleanable oil filter, 4 in. thick, with 24 layers of expanded metal and wire screen graded from coarse mesh at entrance to fine mesh at leaving side.

B-2—A viscous coated throw-away type of filter, 2 in. thick. The fibrous media were graded in fiber size, density, and oiling from entering to leaving side. This filter is designated in this report as *B-2* because slight changes had been made in the oiling

* Director, Engineering Experiment Station, University of Minnesota.
** Instructor, Engineering Experiment Station, University of Minnesota.

[1] ASHVE RESEARCH REPORT No. 1094—Air Filter Performance as Affected by Kind of Dust, Rate of Dust Feed and Air Velocity Through Filter, by Frank B. Rowley and Richard C. Jordan. (ASHVE TRANSACTIONS, Vol. 44, 1938).

[2] ASHVE RESEARCH REPORT No. 1122—Air Filter Performance as Affected by Low Rate of Dust Feed, Various Types of Carbon, and Dust Particle Size and Density, by Frank B. Rowley and Richard C. Jordan. (ASHVE TRANSACTIONS, Vol. 45, 1939).

Presented at the Fall Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Atlanta, Ga., October, 1939.

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and fiber diameter from those filters used in the previous reports. These changes, however, did not appear to make any appreciable difference in the test results.

C—A cellular type filter, 2 in. thick. The filter was built in two sections, each with the axis of cells set at 45 deg to the center line of duct and at 90 deg to each other. The cells on the entering side were of larger dimension than those on the leaving side of the filter.

D—A filter of cotton media of coarse material on the entering side and glazed on the leaving side. The filter media were accordion plaited in frame to give an area of approximately 12 times the cross-sectional area of air stream.

There are two fundamental requirements of any dust that is to be used for filter test work. First, it must simulate air-borne dust in all of its properties

FIG. 1. VIEW OF ASSEMBLED TEST APPARATUS

which may affect filter performance. Second, it must be possible and practical to obtain the dust in sufficient quantities for test purposes and with reasonable certainty that all of the properties can be duplicated.

The properties of a test dust which must be comparable to those of air-borne dust may be summed up as follows:

1. It must be similar in density and air flotation characteristics.
2. It must have reasonable likeness in range and distribution of dust particle size.
3. It must not have a tendency to agglomerate and thus change the physical characteristics of the dust.
4. It must not create any physiological or other hazard.

The standard filter test dust selected for the ASHVE Air Filter Code consists of 50 per cent by weight of powdered lampblack containing a minimum

of 97.5 per cent free carbon and having a minimum bulk density of 3.5 lb per cubic foot, mixed with 50 per cent by weight of Pocahontas bituminous coal ash screened to pass a 200-mesh screen. There are many kinds of dust in the air with wide variations in density, particle size and shape, as well as other characteristics. Previous investigations carried out in this test program have shown that many of these factors have a marked effect on filter performance, and that a given dust may not have a like effect on the performance of two different types of filters. Furthermore, during the early part of the filter research program it was discovered that rather wide variations might be encountered in the results for tests performed on identical filters using dust mixtures of the same specifications. Apparently some of these variations were caused by slight differences in the construction of the filters, but the major

Table 1—Materials Investigated for Artificial Test Dusts

Type of Dust	Reasons for Discarding from Consideration
Fuller's Earth	Absorptive when used in large percentages
Air Floated Silica (325-mesh)	High density—physiological hazard
Borax	Tendency to agglomerate and fall to bottom of test duct
Rosin Dust	Too sticky
Chalk Dust	High density
Alum	High density
Ground English Walnut Shells (parting dust)	Apparently too absorptive—Filter arrestance dropped off rapidly in test
TiO_2	Plugged feeder—Tendency to agglomerate and fall to bottom of test duct
Talc and soapstone parting dust	High density
Lycopodium	Average particle size too large
Al_2O_3	High density
Pumice	High density
Wood Ash	Too difficult to obtain in quantities

variations appeared to be caused by some differences in the components of the dust mixtures. In order to eliminate the effects of variations in the dust constituents throughout any series of tests, it was found necessary to obtain a sufficient amount of each type of dust at the beginning of the series to last throughout all of the tests involved. From this it was evident that the specifications for the synthetic test dust should be determined more rigidly in order to aid duplication, and that possibly the ratios of the constituents should be changed in order to simulate more nearly the actual dust in the air.

The previous investigations showed that in so far as filter performance was concerned there were two general classes of dusts; first, the light flocculent dusts represented by lampblack and carbon black; and second, the more granular dusts represented by the coal ashes. Studies of dust samples taken from the air show reasonable counterparts of these two synthetic constituents.

Before making an extended investigation of various types of carbon and coal ash many other types of dusts which have been suggested and used to some

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TABLE 2—SOURCE OF SAMPLE AND DENSITY OF TEST ASHES
(*Coal burned at uniform rate*)

DISTRICT	COAL SAMPLE	DUST DENSITY IN GRAMS/CC (JOLTED)	SOURCE
West Virginia	A	0.737	Sewell Seam, Fayette County, W. Va.
	B	0.761	Dorothy Seam, Crown City, W. Va.
	C	0.790	No. 3 Seam, McDowell County, W. Va.
	D	0.818	No. 3 Seam, McDowell County, W. Va.
	E	1.028	Beckley Seam, Raleigh County, W. Va.
Illinois	F	0.661	Herrin No. 6 Seam, Perry County, Ill.
	G	0.667	Herrin No. 6 Seam, Franklin County, Ill.
	H	0.722	Harrisburg No. 5 Seam, Saline County, Ill.
	I	0.740	Belleville No. 6 Seam, Macoupin County, Ill.
	J	0.852	Springfield No. 5 Seam, Fulton County, Ill.

extent for air filter tests were considered. Some of the dusts investigated, together with the reasons for not accepting them as satisfactory, are shown in Table 1. While all of the dusts listed in this table were unsatisfactory for the reasons given, it is possible that further investigation may produce a material which will have all of the requirements and be more satisfactory than any combination of carbon and ash. However, as no such material was apparent, an extended investigation was made into the properties of carbon and ash dusts.

Ash

There is a wide variation possible in the chemical and physical properties of coal ash, even though the coal has the same commercial designation. These

TABLE 3—FILTER PERFORMANCE[a] FOR WEST VIRGINIA POCAHONTAS AND ILLINOIS COAL ASHES ON FILTERS *B*-2 AND *C*
(*Dust feed* 40 *g per hour. Face air velocity* 300 *fpm*)

SOURCE	COAL SAMPLE	JOLTED DENSITY GRAMS/CC	FILTER *B*-2			FILTER *C*		
			Arrestance in Per Cent	Life in Hours	Dust Holding Capacity, Grams	Arrestance in Per Cent	Life in Hours	Dust Holding Capacity, Grams
West Virginia	E	1.028	90.8	11.0	400.1	....	....	....
	D	0.818	94.2	7.3	275.1	85.5	9.0	307.7
	C	0.790	94.8	7.8	295.3	83.7	9.0	301.2
	B	0.761	93.2	6.2	233.3	78.3	8.4	262.5
	A	0.737	94.8	6.8	258.0	82.4	9.5	313.1
Illinois	J	0.852	95.4	9.3	355.4	82.9	10.9	362.0
	I	0.740	96.6	6.6	255.4	86.7	11.3	394.9
	H	0.722	96.0	6.3	241.9	....	....	....
	G	0.667	95.6	7.1	271.0	84.6	8.7	294.9
	F	0.661	94.4	5.0	222.7	83.4	8.8	293.6

[a] Based on resistance rise of 0.15 in. of water across filter.

differences may be inherent in the original coal or may be due to some element in the combustion process by which the ash is formed. In order to investigate the effect of some of these variations on filter performance, samples were selected of coal from five different West Virginia Pocahontas fields and five different Illinois coal fields. All of these coals were burned at uniform rates of combustion and one of the Pocahontas samples was also burned at different rates of combustion. The resulting ashes were used for tests on two different types of filters.

In making these tests the jolted density of the resulting ash was determined by placing the dust in a cylindrical container $1\frac{7}{8}$ in. high and $2\frac{21}{32}$ in. in diameter, and tapping to maximum density. The source of the sample and

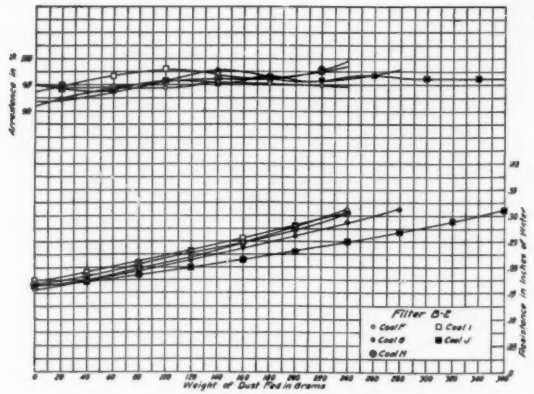

FIG. 2. EFFECT OF FIVE DIFFERENT ILLINOIS COAL ASHES ON PERFORMANCE OF FILTER *B-2*

densities of the resulting ash when burned at a uniform, moderate rate are shown in Table 2 and the densities of the ash for slow burning as compared with fast burning are shown for Sample *C* in Table 7. The moderate rate of combustion was estimated as that corresponding to the average hand-fired domestic furnace rather than the average stoker-fired furnace.

The various test dusts as shown in Tables 2 and 7 were used for testing both Filters *B-2* and *C*. In these tests the face air velocity was maintained at 300 fpm and the ash was fed to the filter at the rate of 40 g (grams) per hour. Each test was continued until the pressure drop across the filter had risen to 0.15 in. of water. The results of these tests are shown in Tables 3 and 7. The complete graphical test results for Filter *B-2* on the Illinois coal ashes at a moderate rate of combustion are shown in Fig. 2. These curves are typical for all of the tests.

The test results of Table 3 indicate that for Filter *B-2* the arrestance values are in close agreement for the different densities of each coal with the exception

of the Pocahontas coal, *E*, which has a very high density and shows a somewhat lower arrestance value for this filter. For Filter *C* the arrestance values are not as consistent on either type of ash, although the variations are not serious. No doubt there is some difference in the particle size for the different ash samples as well as differences in density which will affect the arrestance values. There is a rather definite linear relationship between dust density and filter life for both the Pocahontas and Illinois coal ashes for both filters, as indicated in the curves of Fig. 3.

Since different filters were used with the different samples of ash it is likely that there would be some variation in the characteristics of the filters even though they were of the same make. In order to determine the effect of this variation, tests were made on five different filters of each type using Pocahontas ash from the same lot. The results for these tests as shown in Table 4 indicate a moderate variation between the different filters of the same type.

The figures of Table 5 give the variations in performance characteristics

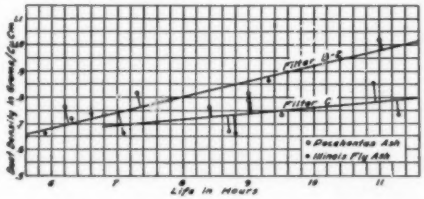

FIG. 3. EFFECT OF ASH DUST DENSITY ON FILTER LIFE. DUST FEED, 40 G PER HOUR

for Filters *B-2* and *C* for ash densities between 0.65 and 0.85 g per cubic centimeter. As will be noted the range in arrestance values is very small. For Filter *B-2* the percentage variation from the average for arrestance, life and dust holding capacity are substantially the same when considering either the Pocahontas ashes or the Illinois ashes as a group, or the single Pocahontas ash. For Filter *C* the variations are greater for the group than for the single ash.

An indication of the effect of ash variation alone on filter performance is given in Table 6. The values shown in this table were obtained from those in Table 5 by subtracting the range in performance units for one ash from the range in performance units for various ashes and correcting the percentages accordingly. The results of this table are not all based on the same number of tests. The negative values may be considered within the range of experimental error.

The values in Tables 5 and 6 give an indication of the magnitude of the error which is likely to be introduced by the use of various ashes. They do not indicate true ranges of percentages as they are not based upon a sufficient number of tests for statistical accuracy. The running of a sufficient number of tests to accomplish this would have involved too much time and expense and probably would have added little to the results. The tabulations in Table 6

indicate the greatest variations in performance characteristics to be for Filter *C*. This probably is due to the fact that variations in the average dust particle size have more effect on Filter *C* than on any of the other filters tested.

The test results for Filters *B-2* and *C* on the ashes from one Pocahontas coal burned at two different combustion rates are shown in Table 7. The slow rate of combustion was comparable with that of a well banked domestic fire, while the fast rate of combustion was somewhat in excess of that encountered in domestic installations. The ash from the fast burned coal had the highest density and, on filters, resulted in higher arrestance, longer life, and greater dust holding capacity than the slow burned ash. In the fast burned coal the lighter fractions of the ash probably were carried off with the flue gases, thus giving a denser ash of greater particle size. The greater average particle size would explain the higher arrestance, and the higher density the increased life

TABLE 4—FILTER PERFORMANCE[a] VARIATION FOR FILTERS *B-2* AND *C* WITH SAME LOT OF POCAHONTAS ASH
(*Dust feed* 40 *g per hour. Face air velocity* 300 *fpm*)

FILTER	TEST NO.	PERFORMANCE CHARACTERISTICS		
		Arrestance in Per Cent	Life in Hours	Dust Holding Capacity, Grams
B-2	1	93.2	7.0	261.2
	2	93.7	6.7	251.3
	3	94.8	7.8	295.3
	4	93.2	6.5	242.1
	5	93.2	7.0	261.2
C	1	82.1	9.7	319.6
	2	83.6	8.9	298.5
	3	83.7	9.0	301.2
	4	83.7	9.1	305.4
	5	81.7	9.2	299.6

[a] Based on resistance rise of 0.15 in. of water across filter.

and dust holding capacity. These tests indicate that any standards set up for the ash constituent of a dust mixture should include a limiting maximum combustion rate.

Carbon

In a previous investigation carried out in the general filter program four types of carbons (lampblack, carbon black, bone black, and soot) were selected and a study made of the effects of these materials on the performance characteristics of Filters *A, B-1, C,* and *D*. The results of this investigation showed that a wide range of results could be obtained dependent upon the particular carbon used, and that either lampblack or carbon black would make a satisfactory carbon ingredient for a standard dust if it could be duplicated.

In order to determine the variations in test results which might be expected from various carbon materials which would come within the code specifications, samples were selected of both lampblack and carbon black. (*Note:* the distinc-

TABLE 5—FILTER PERFORMANCE[a] VARIATION WITH DIFFERENT ASHES
(*Ash Density* 0.65 *to* 0.85 *g per cc*)

FILTER	CHARACTERISTICS	RANGE FOR VARIOUS POCAHONTAS ASH		RANGE FOR VARIOUS ILLINOIS ASH		RANGE FOR ONE POCAHONTAS ASH	
		Range in Performance Units	$\frac{\text{Range}}{\text{Avg}} \times 100 =$	Range in Performance Units	$\frac{\text{Range}}{\text{Avg}} \times 100 =$	Range in Performance Units	$\frac{\text{Range}}{\text{Avg}} \times 100 =$
B-2	Arrestance	1.6%	1.7%	2.0%	2.1%	1.6%	1.7%
	Life	1.6 hours	22.9%	1.2 hours	18.5%	1.3 hours	18.6%
	Dust Holding Capacity	69.5 g	26.0%	48.3 g	19.5%	53.2 g	20.5%
C	Arrestance	7.2%	8.8%	3.3%	3.9%	2.0%	2.4%
	Life	1.1 hours	12.3%	2.6 hours	27.1%	0.8 hours	8.7%
	Dust Holding Capacity	50.6 g	17.1%	101.3 g	30.9%	21.1 g	6.9%

[a] Based on 0.15 in. resistance rise; face air velocity 300 fpm; dust feed 40 g per hour.

TABLE 6—FILTER PERFORMANCE[a] VARIATION WITH DIFFERENT ASHES CORRECTED FOR FILTER VARIATION

(*Ash density* 0.65 *to* 0.85 *g per cc*)

FILTER	PERFORMANCE CHARACTERISTIC	RANGE FOR VARIOUS POCAHONTAS ASHES		RANGE FOR VARIOUS ILLINOIS ASHES	
		Range in Performance Units	$\frac{\text{Range}}{\text{Avg}} \times 100 =$	Range in Performance Units	$\frac{\text{Range}}{\text{Avg}} \times 100 =$
B-2	Arrestance	0.0%	0.0%	0.4%	0.4%
	Life	0.3 hours	4.3%	−0.1 hours	− 1.5%
	Dust Holding Capacity	16.3 g	6.1%	−4.9 g	− 1.9%
C	Arrestance	5.2%	6.4%	1.3%	1.5%
	Life	0.3 hours	3.4%	1.8 hours	18.8%
	Dust Holding Capacity	29.5 g	10.0%	80.2 g	24.4%

[a] Based on 0.15 in. of water resistance rise; face air velocity 300 fpm; dust feed 40 g per hour.

tion between the two types of materials is that lampblack is formed by burning of oil, and carbon black by burning of gas.) All of the samples of each of these materials were of the same brand but selected from different lots and from material manufactured at different dates.

The test results from the different samples of lampblack are shown in Table 8. In this series of tests the face air velocity was maintained at 300 fpm with a dust feed of 40 g per hour. The same types of filters, *F-1*, were used in all tests. As it will be noted from the test results, there was a spread in arrestance values from 28.1 to 82.4. The lowest arrestance values were from lot *A*, and the 28 per cent arrestance was obtained on one of the earlier samples selected. There was a spread in the arrestance values for the samples of lot *B* from 56.8 to 82.4, and a spread in the values for life of filter from 1.5 to 4.5 hours. The significant point in these test results is the wide spread obtained in arrestance values and the uncertainty of consistent results from

TABLE 7—EFFECT OF ASH BURNING RATE ON PERFORMANCE[a] OF FILTERS *B-2* AND *C*

(*Face air velocity* 300 *fpm. Dust feed* 40 *g per hour*)

FILTER	ASH	DENSITY (JOLTED) IN GRAMS PER CC	ARRESTANCE PER CENT	LIFE IN HOURS	DUST HOLDING CAPACITY, GRAMS
B-2	No. 3 McDowell City Pocahontas (fast burned)	0.860	94.3	7.5	282.8
B-2	No. 3 McDowell City Pocahontas (slow burned)	0.765	90.2	5.5	198.2
C	No. 3 McDowell City Pocahontas (fast burned)	0.860	85.2	10.7	365.0
C	No. 3 McDowell City Pocahontas (slow burned)	0.765	78.9	7.9	249.5

[a] Based on 0.15 in. of water resistance rise; face air velocity 300 fpm; dust feed 40 g per hour.

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different samples of this material. It would be impractical to use a material giving such a wide range of test results as a basis for a synthetic test dust.

In making the tests on carbon black, two different brands of material were selected. These were known as Double-Bolted Carbon Dust and Super-Spectra Carbon Black, both obtained from Binney and Smith Co., New York City. Six samples were selected from the first material and three from the second, all taken from different shipments manufactured at different times. In making these tests Filter *B-2* was used, the face velocity was maintained at 300 fpm and the dust feed at 20 g per hour. The rate of dust feed was set at 20 g as against 40 g for the lampblack because of the lower density of the carbon black. The test results on both brands of carbon black as given in Table 9

TABLE 8—COMPARISON OF LAMPBLACKS[a] ON FILTER *B-1*
(*Face air velocity* 300 *fpm. Dust feed* 40 *g per hour*)

SOURCE	DUST SAMPLE	DUST DENSITY	AVG RESIS.[b] RISE PER HUOR	LIFE[c] IN HOURS	AVG ARRES. FOR 2 HOURS
A	1	0.156	0.024	Approx. 4.5	28.1
	2	0.238	0.066	2.3	46.7
B	1	0.183	0.088	1.8	78.8
	2	0.213	0.089	1.8	74.0
	3	0.203	0.084	1.9	78.4
	4	0.206	0.063	2.3	75.6
	5	Approx. same as 1, 2	0.087	1.9	82.4
	6	Approx. same as 1, 2	0.084	1.9	64.4
	7	Approx. same as 1, 2	....	....	56.8
	8	Approx. same as 1, 2	0.098	1.8	66.2
C	1	0.209	0.074	2.1	71.1
D	1	0.265	0.049	1.5	58.4

[a] All lampblacks same brand and same manufacturer but from different supply houses.
[b] Based on 2 hour average.
[c] Based on final resistance of 0.30 in. of water across filter.

show a much higher degree of uniformity than those for lampblack. In the case of Super-Spectra Carbon Black there was a marked drop in arrestance after the second hour of the test, which was probably caused by the inability of the oil to absorb the dust as rapidly as it was fed. Even though the rate was only 20 g per hour the extremely low density of this dust resulted in a high volumetric rate of feed. Of the various carbon dusts tested the Double-Bolted Carbon Dust appears to have the best properties for a dust to be selected as a standard to fulfill the carbon requirements in the code.

It should be noted that the requirements of a carbon dust for filter testing are very exacting and may be entirely different than the requirements of the dust when used for commercial purposes.

Dust Mixture

From an analysis of dust samples taken from the air it appears that there are two typical types of dust, and from a study of many types of artificial

dusts it appears that these two natural dusts in the air may be simulated with carbon black and Pocahontas ash, each prepared by a definite process. Tests have shown the performance characteristics of the various filters to be quite different when using the two different types of dust. Thus it appears that it is not necessary to use more than two different types of dust in a test mixture, but that it is important that these dusts be mixed in substantially the same proportions as they will be found in the air which must be handled by the filter.

Most of the test dust mixtures used at the present time are proportioned on a weight basis. Since there is a wide variation in the densities of the individual dusts, there will be a big difference in the ratio of the components when calculated on a volume basis. The figures in Table 10 show the analysis of six different test dusts both on the weight and volumetric basis. The figures in the second column show the wide variation in densities of the different

TABLE 9—COMPARISON OF FILTER *B-2* PERFORMANCE CHARACTERISTICS ON TWO CARBON BLACKS

(*Face air velocity* 300 *fpm. Dust feed* 20 *g per hour*)

BRAND CARBON BLACK	SAMPLE NO.	DENSITY IN GRAMS PER CC (JOLTED)	RESISTANCE RISE IN 3 HOURS (IN. OF H_2O)	AVG ARRESTANCE
Doubled Bolted Carbon Dust	1	0.154	0.118	52.5
	2	0.148	0.125	53.2
	3	0.162	0.130	59.6
	4	0.155	0.135	51.0
	5	0.171	0.123	54.7
	6	0.159	0.126	53.1
	7	0.177	0.143	59.2
Super-Spectra	1	0.066	0.152	64.5
	2	0.068	0.129	68.0
	3	0.068	0.131	67.7

mixtures, and the figures for the percentages by weight and volume of the different dusts in the mixture show the wide variation in the components on the weight and volume basis. The sample made up of 50 per cent Pocahontas ash and 50 per cent lampblack as specified by the standard test code shows only 17 per cent of Pocahontas ash by volume and 83 per cent of lampblack. The density of the combined sample is less than one-third that of Cottrell ash, which is sometimes used as a test dust.

Although air-borne dust varies greatly in composition, depending upon such factors as locality, weather, time of year, etc., it seems improbable that it would ordinarily consist of more than 50 per cent by volume of carbon. This fact was recognized by manufacturers and users of filters when a mixture of 50 per cent Pocahontas ash, 20 per cent Illinois fly ash, 20 per cent lampblack, and 10 per cent Fuller's Earth was selected as a basis for railroad air filter tests made at the University of Minnesota in 1937. Usually lampblack or carbon black is a very difficult dust to separate out of the air, and most filters which will successfully separate it show short life and low dust holding capacity. If a test dust mixture is to contain 83 per cent by volume of a dust which has

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very decisive effects on filter performance and which is found in much lower percentages in average air, it is evident that the rating of the filter will be misleading.

It might be advisable to formulate more than one standard dust mixture for the purpose of rating filters to meet the requirements of the dust load which they must handle. Dust filters to be operated in smoky areas would be tested with a dust mixture incorporating a higher percentage of carbon than filters to be operated in areas relatively free from smoke. For the determination of the general performance of filters it would appear advisable to have a basic standard mixture, the alternate mixtures to be used only in case the manufacturer or purchaser desires ratings for specific purposes. The base mixture

TABLE 10—VOLUMETRIC ANALYSIS OF DUST MIXTURES BY WEIGHT

DUST MIXTURE PER CENT BY WEIGHT	VOL. IN CC OF 40 G OF MIXTURE	VOL. PER CENT LAMPBLACK	VOL. PER CENT CARBON DUST	VOL. PER CENT POCAHONTAS ASH	VOL. PER CENT ILLINOIS ASH	VOL. PER CENT FULLER'S EARTH	VOL. PER CENT COTTRELL ASH
50% Pocahontas Ash 20% Illinois Ash 20% Lampblack 10% Fuller's Earth	95.0	54.0	...	27.5	11.3	7.2	...
50% Pocahontas Ash 50% Lampblack	154.3	83.0	...	17.0	...	...	...
90% Pocahontas Ash 10% Carbon Dust	72.2	...	35.0	65.0	...	...	...
80% Pocahontas Ash 20% Carbon Dust	92.4	...	54.8	45.2	...	...	...
70% Pocahontas Ash 30% Carbon Dust	112.5	...	67.5	32.5	...	...	...
100% Cottrell Ash	31.1	...	...	...	...	...	100.0

would be as representative as possible of average air-borne dust and result in a reasonable range in performance characteristics for the different filters. A dust mixture which shows a high arrestance on the average filter shows little range between the arrestance values for different filters. The opposite is true of dust resulting in a comparatively low arrestance on average filters. Furthermore, for the same weight of dust feed per hour, a low density dust will result in short filter life and low dust holding capacities, while a higher density dust will show the other extreme. A basic standard dust mixture should show none of these extremes.

As previously explained, the investigations made on the various types of dusts indicate that the required elements of a dust mixture can be supplied by the two dusts, Pocahontas ash and carbon black. These two types of dust were

Table 11—Comparison of Filter *B-2* Performance Characteristics[a] on Three Pocahontas Ash-Carbon Dust Mixtures
(*Face air velocity* 300 *fpm. Rate of dust feed* 40 *g per hour*)

Mixture	Avg Arrestance, Per Cent	Life in Hours	Dust Holding Capacity, Grams
70% Pocahontas Ash 30% Carbon Dust	84.4	3.5	117.4
80% Pocahontas Ash 20% Carbon Dust	87.8	4.0	140.7
90% Pocahontas Ash 10% Carbon Dust	90.9	5.0	182.0

[a] Based on final resistance of 0.30 in. of water across filter.

thus selected and mixed in different ratios in order to find a suitable mixture as a basic test dust. The dusts were mixed in percentages by weight of 90-10, 80-20, 70-30 of Pocahontas ash and carbon dust respectively. These dust mixtures were used with Filter *B-2* at a face velocity of 300 fpm and a rate of dust feed of 40 g per hour. From the results of these tests shown in Table 11 it was decided that the 80-20 ratio, giving approximately 50 per cent by volume of each type of dust, was the most representative of the three mixtures for a base test dust.

The 80-20 dust mixture was used for testing Filters *A*, *B-2*, *C*, and *D*, and the resulting characteristics compared with test results for the 50-50, 50-20-20-10, and the 100 per cent Cottrell ash tests. The results of these tests are shown in Table 12. A typical set of performance curves for the 50-50, and

Table 12—Comparison of Filter Performance[a] on Four Test Dusts
(*Face air velocity* 300 *fpm. Dust feed* 40 *g per hour*)

Type Filter	50–50 Mixture			50–20–20–10 Mixture		
	Arr.	Life	D. H. C.	Arr.	Life	D. H. C.
Filter A	60.9%	9.0	219.1	79.9%	12.0	383.2
Filter B-2	....	....	....	84.7%	3.0	101.7
Filter C	59.0%	3.9	91.5	73.6%	6.0	175.0
Filter D	92.7%	2.2	81.6	96.2%	4.9	188.7

Type Filter	80–20 Mixture			Cottrell		
	Arr.	Life	D. H. C.	Arr.	Life	D. H. C.
Filter A	79.1%	10.0	316.3	92.3%	36.0	1329.0
Filter B-2	87.8%	4.0	140.7	....	....	....
Filter C	70.1%	6.8	191.0	78.4%	22.8	715.0
Filter D	94.4%	6.9	260.5	88.1%	34.0	1198.0

[a] Based on final resistance of 0.30 in. of water across filter.

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50-20-20-10 mixtures is shown in Fig. 4 for Filter *A*, and the performance characteristics of the 80-20 mixture for Filters *A*, *B-2*, *C*, and *D* are shown in Fig. 5. The 50-50 mixture resulted in the lowest arrestance, shortest life and dust holding capacity, while the Cottrell ash showed the other extremes. The 50-20-20-10 mixture gave approximately the same results as the 80-20 mixture, probably because each contained 20 per cent by weight of carbon, and because Fuller's Earth and Pocahontas ash both have about the same effect on filter performance. The test values on these two mixtures lay between those for the 50-50 and the Cottrell test dusts.

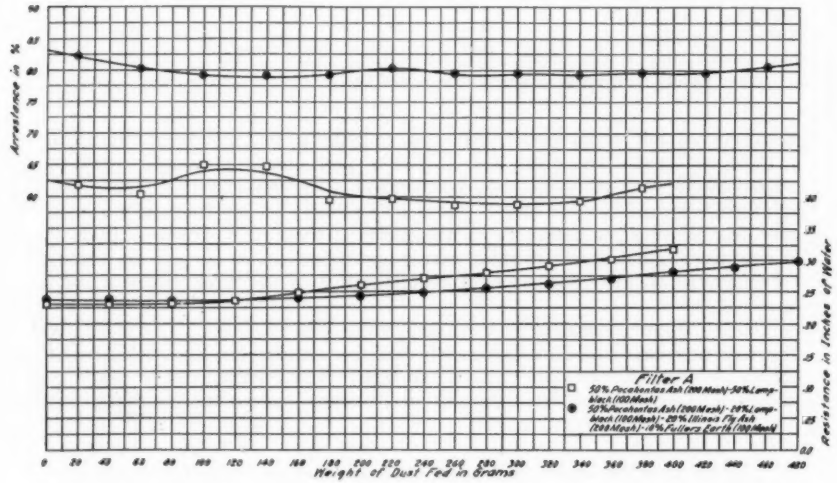
Arrestance in %
Resistance in Inches of Water
Weight of Dust Fed in Grams
Filter A
50% Pocahontas Ash (200 Mesh)-50% Lampblack (100 Mesh)
50% Pocahontas Ash (200 Mesh)-10% Lampblack (100 Mesh)-20% Illinois Fly Ash (200 Mesh)-10% Fullers Earth (100 Mesh)

FIG. 4. PERFORMANCE CURVES FOR FILTER *A* ON TWO DUST MIXTURES. DUST FEED, 40 G PER HOUR

A test dust such as Cottrell ash results in performance characteristics which are probably better than those actually obtained in practice, whereas a dust such as the 50-50 mixture probably penalizes the filter unduly. The 80-20 and the 50-20-20-10 mixtures give approximately the same results, but the former is easier to prepare because it has only two components.

CONCLUSIONS

There are two general classes of dusts in so far as their effects on the rating of air filters on a weight basis are concerned; first, the light flocculent dusts represented by lampblack and carbon black; and second, the more granular dusts represented by the coal ashes. These dusts vary in density and particle size and it is necessary to control these variations within reasonable limits if the effects of the dusts on filter performance are to be capable of duplication. Numerous tests conducted with various ashes and carbons indicate that carbon black and Pocahontas ash satisfy the requirements of a standard test dust if controlled within reasonable

limits as to density and rate of burning. Changes in manufacturing processes may affect the standardization of the carbon black, but for any given set of manufacturing conditions there was less variation in the carbon blacks than the lampblacks tested. Fifty per cent by weight of lampblack as specified in the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work results in 83 per cent by volume of lampblack. This is apparently in excess of the amount of carbon ordinarily found in the air, and 20 per cent by weight of carbon appears to be a more reasonable specification. For these reasons the mixture of 80 per cent by weight of Pocahontas ash screened through a 200 mesh sieve and having a jolted density between 0.65 and 0.85 g per cubic centimeter, mixed with 20 per cent by weight of Double-Bolted Carbon Dust

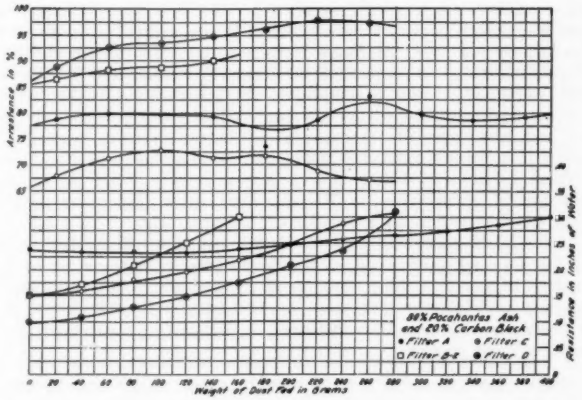

FIG. 5. PERFORMANCE CURVES FOR FILTERS *A*, *B-2*, *C*, AND *D* ON 80-20 DUST MIXTURE. DUST FEED, 40 G PER HOUR

screened through a 100 mesh screen and having a jolted density between 0.145 and 0.180 g per cubic centimeter is recommended as a standard test dust to be used for the general rating of air filters on a weight basis in the laboratory.

DISCUSSION

ARTHUR NUTTING[2] (WRITTEN): This is the third report of air filter research by these authors since August 1938 and I am certain that the great fund of information contained therein has been of real value to all air filter manufacturers. The relationship of air filter performance to dust characteristics, as brought out by these researches, is in itself a basis for a treatise on air cleaning—the lack of which is a serious handicap in many critical air conditioning problems.

Before discussing this paper, it is well to point out that filter manufacturers and air conditioning engineers are interested in the specification of a standard testing dust because the dust, more than any other factor, affects the ratings given the filter. This is conveniently illustrated in Table 12 by the excessively high dust holding capacities and arrestance of the filters when tested with Cottrell dust. Filter *D* for instance held 15 times as much Cottrell as it held of the ASHVE 50-50 mixture.

[2] Chief Engineer, American Air Filter Co., Inc., Louisville, Ky.

Each of these reports has been summarized with the remark that the present ASHVE Code dust, which contains a 50-50 proportion of ash and lampblack, handicaps the air filter performance, that is, gives low test values. In view of this the authors suggest that the per cent of free carbon in the code dust should be reduced to the proportion usually found in the atmosphere. They reach this conclusion after comparing their filter efficiency values on the standard dust with values on other artificial dusts they use, but nowhere are there any comparisons made with air filter performance under actual field conditions. It seems entirely illogical to argue at this time that one artificial dust gives truer values than another, when there are no data on the real values of filter performance obtained in practice.

Years ago, the ASHVE mixture was adopted after serious consideration. At that time all the committee knew about air filter performance was that some were good cleaners and some were poor. They set out to select an artificial dust which would for the best and poorest types of filters produce a range of filter performance in the laboratory equivalent to the range of performance experienced in the field. The present ASHVE standard dust accomplishes this purpose remarkably well, and it is by no means too rigid, for it gives an efficiency difference of only 36 per cent between the poorest and best types of filters as given in the previous paper.[4] The 80-20 mixture which is being recommended at present, reduces this difference to 24 per cent between Filters *C* and *D*. Filter *C* the poorest on the ASHVE mixture gained 11 per cent (from 59 efficiency to 70 efficiency) by a change to the 80-20 dust. Filter *D*, an excellent filter on the ASHVE mixture, gained only 1.7 per cent by changing to the 80-20 mixture. It is evident from chart 12 of the present report that the performance of filters *A*, *B*, *C* are raised entirely at the expense of Filter *D*. If the tendency is to select dusts easy to catch, and thus bring all types of filters within a narrow performance range, no one will be able to distinguish, by test, the poor filter from a good one. Manufacturers of superior equipment will be penalized and engineers will not be able to predict true performance.

The argument that any artificial dust should have carbon content equal to atmospheric dust is not entirely reasonable. It is necessary to have an excessive amount of small size carbon particles in the present standard dust to balance the large size Pocahontas ash particles, which are much larger than atmospheric particulate matter. It is impossible to produce, in large quantities, a granular dust having a size distribution approaching atmospheric dust sizes, and the fact that the ash constituent of an artificial dust must necessarily be extremely large is a strong argument against the use of as much as 80 per cent of it in the test code. Furthermore, the free carbon in atmospheric solids creates the real and almost the entire dust nuisance. It is the free carbon that causes practically all of the soiling and discoloration that the purchasers of air filters are trying to eliminate. Evidence of its presence in an improperly filtered air system is entirely out of proportion to the 20 per cent value the authors place on it. An air filter, which removes all impurities except carbon, serves no useful function in the ventilating system.

This report suggests that the manufacturers and users of filters recognized a true dust when the mixture of 50 per cent Pocahontas ash, 20 per cent Illinois fly ash, 20 per cent lampblack and 10 per cent Fuller's Earth was selected for the railroad air filter test made at the University of Minnesota in 1937. This railroad testing dust has 80 per cent granular material like that being suggested now. I would like to point out that this mixture was selected to correspond with the dust picked up by filters in a train operating between the Pacific Coast and Chicago. The dust storms had made the railroads air filter conscious and the 80 per cent of large size granular particles in their test dust are supposed to be representative of the sand and clay blown about the prairies. The fact that 80 per cent granular matter was then

[4] Loc Cit. Note 2.

selected for a *country* dust would indicate that a *city* dust, which is entirely different in character, should have far less then 80 per cent granular.

I would like to point to one item of interest in Table 12. Filter *B* is the only filter which shows a higher arrestance on the proposed 80-20 dust than on the 50-20-20-10. All the other filters show a lower arrestance for this dust, in fact Filters *C* and *D* each lose as much as Filter *B* gains. It is difficult to reconcile the selection of a code dust which would discriminate in favor of one device and against all the others.

The conclusions of this report are based on the performance of viscous unit Filters *A*, *B* and *C*. In arriving at these conclusions no consideration is given to the fabric or dry type of filter designated as *D*. In comparison to the other filters, this type is superior on the ASHVE dust. On the proposed 80-20 mixture this superiority, in relation to the other types, is lost and yet, this type filter employing paper, wool, felt, cotton, etc., represents one of the most commonly used filter types where good air cleaning performance is required. Disregard of this or any other widely used kind of filter, in any code dust specification, would indeed be a serious blunder.

Today, the trend is toward a higher degree of air filter performance. In the past few years, air filter design and performance has undergone more revolutionary changes than perhaps any other air conditioning device. In fact, electrostatic air filters clean the air to such a high degree that more fine carbon particles will be needed in the Code dust to give a true comparative test when this type of filter comes into more general use. It is practically impossible to collect a weighable amount of dust in air that has been cleaned by this type of filter. With electrostatic air filters, the measurement of filtering efficiency is based upon the capacity of the cleaned air to soil or discolor a white surface. In this kind of test, the smallest particles, because of their large surface, cause far more discoloration per unit weight than the larger particles and carbon is the chief constituent of the colored matter.

In view of these remarkable improvements in air cleaning methods, it would seem fair that the test code recognize this higher degree of air cleaning, which has never before been attained. The use of any easy-to-catch dust as a yardstick of filter performance would, by the fictitiously high value given poor type filter, limit unfairly the ratings applied to filters of vastly improved design.

In conclusion, I would like to suggest that this Society prohibit use of its name on published data which have not been made in strict accordance with its Code. Several manufacturers publish filter ratings which are based on the railroad dust and one uses a dust of his own concoction, composed of street sweepings and vacuum cleaner dust. In all instances, the ratings are claimed to have been made under the ASHVE Code.

PROF. F. B. ROWLEY: The main point in Mr. Nutting's discussion appears to be the fact that he is opposed to making a change in the Standard Code dust to reduce the percentage of carbon below 50 per cent. First, I would like to call attention to the fact that while this recommendation has been made and the authors believe it can be substantiated, the percentage of carbon used in the standard dust was not the main point under consideration throughout this investigation. The principal purpose of this investigation was to make a study of different types of carbon and different types of granular dust and to select component materials for the dust mixture which may be duplicated with a reasonable degree of accuracy. As an example, two identical filters tested with two dust mixtures consisting of 50 per cent lampblack and 50 per cent Pocahontas ash which conformed to the Code specifications gave efficiencies varying from 59.5 per cent to 88.6 per cent on Filter *B*. This is a spread of 29.1 per cent units for the same type of filter and for test dusts which comply in every way with the Code. This difference is typical of many other tests and emphasizes the necessity of using standard components which can be duplicated.

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While there is a difference of opinion as to the percentage of each dust which should constitute the standard mixture, one should consider the density as well as the weight of the different types of dust selected. By referring to Table 10 it will be noted that with 50 per cent Pocahontas ash and 50 per cent lampblack by weight as specified by the Code the volumetric percentage is 83 per cent of lampblack to 17 per cent of Pocahontas ash. This would seem entirely out of proportion for any dust which would be found in average air to be handled by an air filter, particularly in a domestic installation. With the 20 per cent lampblack by weight the volumetric percentage is still 54 per cent.

Mr. Nutting has referred to the selection of dust as used for the railroad air filter tests made at the University of Minnesota in 1937, and states that it was selected to represent a country dust rather than a city dust. As a matter of fact, the selection was made by a conference consisting of representatives from all interested filter manufacturers, railroad operators and engineers. The consensus of opinion was that 50 per cent lampblack was entirely too much and the 20 per cent was selected as more nearly representing that which would have to be handled by a railroad air filter, taking into consideration the smoke and carbon from the engine as well as the other dust picked up along the track. In many cases such filters might be subjected to much greater amounts of carbon than filters located in city air.

It should be noted that the authors have recommended not only one type of dust mixture but possibly several alternate mixtures to take into consideration varying filter requirements.

The authors are entirely in accord with the suggestion that filter ratings published as obtained under the Code should comply with the provisions of the Code. However, there are many requirements in the present Code that are not followed by any manufacturers and in some instances requirements which cannot be followed with any degree of uniformity. We believe a person with laboratory experience in the testing of filters will not challenge the statement that two identical filters each tested in accordance with the Code may show widely different results due primarily to the uncertainty in regard to dust mixture used.

In Memoriam

Name	Joined the Society	Died in 1939
Alphonse A. Adler	1921	Jan. 13
Claude A. Bulkeley	1923	July 9
Charles R. Bulleit	1930	Jan. 25
Virgil H. Cherry	1937	June
Edwin C. Hinkle (*Life Member*)	1911	
Richard A. Jennings	1937	Apr. 11
Joaquin G. Jimenez	1935	Spanish Revolution
Robert E. Keyes	1913	May 29
James R. Knox	1930	Oct. 18
W. J. Kyle	1935	Jan. 6
Robert Mautsch	1928	Oct. 19
John F. G. Miller	1916	Dec. 9
James K. Peacock (*Life Member*)	1921	Mar. 7
Herbert A. Ruff	1938	Nov. 16
H. J. Ryan	1922	June 13
Charles Sonneborn	1930	Nov. 4
Theodore J. Steiner	1938	Dec.
Hendrik L. Teves	1938	June 12
George J. Tobin (*Life Member*)	1905	Feb. 14
Michael A. Welter	1925	Dec. 14 *
Bruce M. Wigle	1926	Dec. 6

* Died in 1938.

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